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June 1997
Vol. 18, No.2

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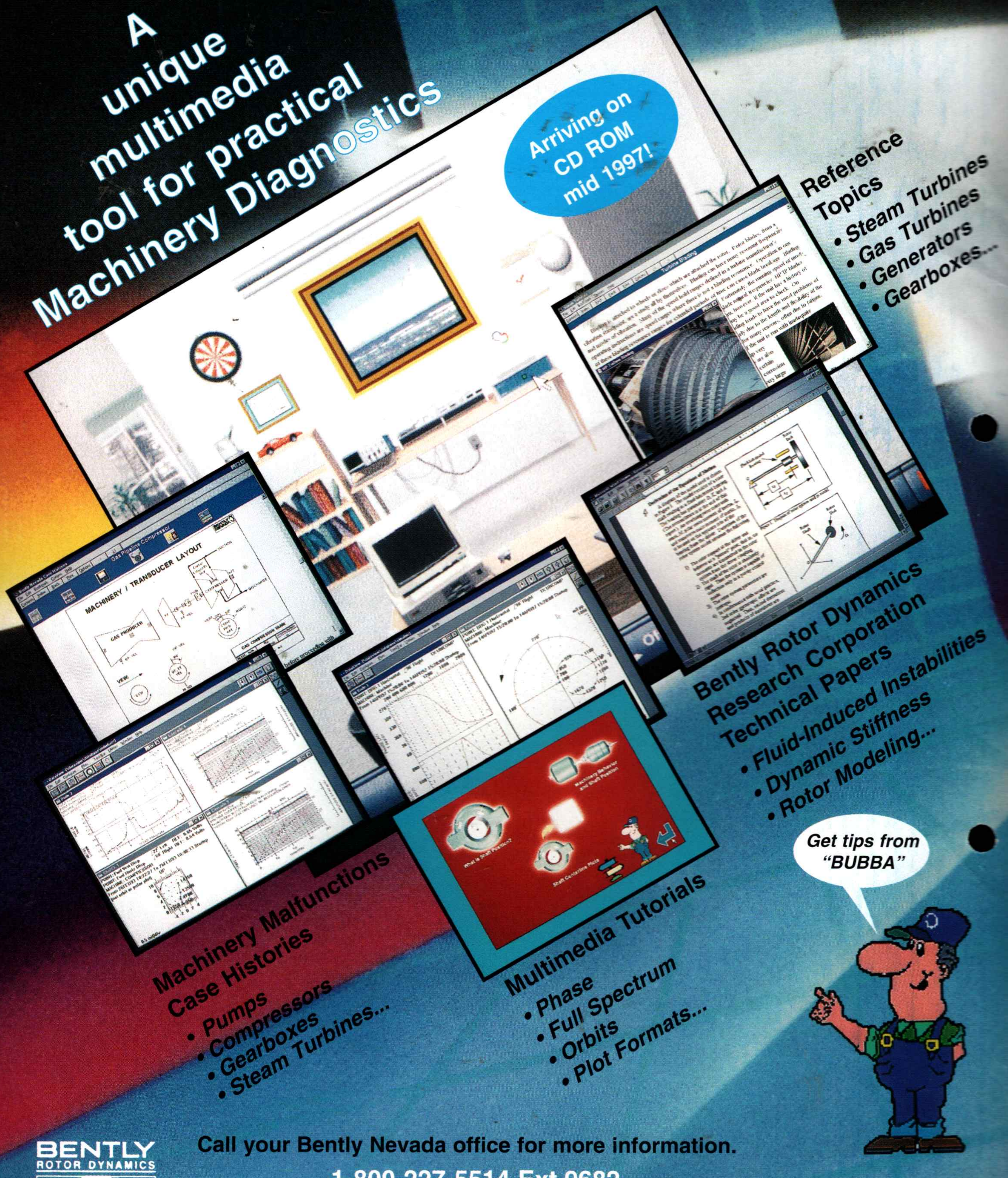
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Printed in USA
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Orbit is published four times per year by Bently Nevada Corporation. Questions, suggestions, and letters to the editor may be addressed to **Orbit**, 1617 Water Street, Minden, Nevada 89423, or contact your nearest Bently Nevada sales representative. Phone: (702) 782-3611 • Fax: (702) 782-9337 • E-mail: cac@bnc73cac.bently.com

Publisher: **Bently Nevada Corporation**
Vice President Sales: **Steven W. Riggs**
Orbit Editor: **Mary Sue Matthews**
Technical Editor: **Bob Grissom**
Art director: **Kenneth R. Winter**
Typographer: **Michael K. Riggs**
Customer Information Services Manager: **Malcom Werner**
Technical Writer: **Dave Burba**
Advertising/Photography: **David Webb**
Technical Artist: **Jeanie Nelson**
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Circulation: **Clydie Costa**

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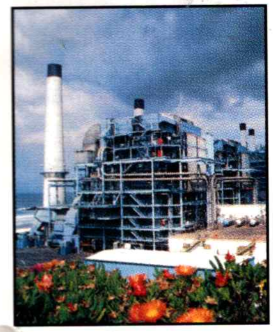
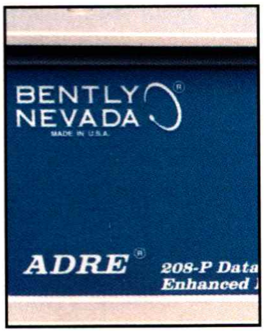


Cover: Southern California Edison's El Segundo Power plant. Bently Nevada's System Integration Services was instrumental in helping them maximize their resources using Data Manager® 2000. See article on page 20.

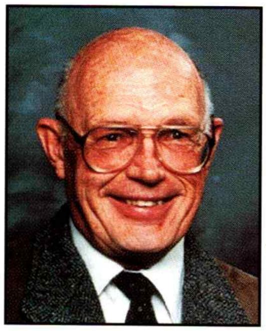
Page 22: Bently Nevada's Transient Data Manager® 2 provided engineers at a Brazilian petrochemical facility appropriate and accurate information. They were able to successfully diagnose a 60% fatigue crack in a coupling.



Page 30: ADRE® for Windows is now more versatile and powerful. Among the improvements are flexible vector math calculations and easy plot rescaling. It also works with a wider variety of notebook and other computers.

From the desk of...

A turbine that succeeds in stabilizing itself



Donald E. Bently
Chairman and Chief Executive officer,
Bently Nevada Corporation
President, Bently Rotor Dynamics
Research Corporation

Nearly all instances of the Fluid-Induced Instability Mechanism (Oil Whip, Steam Whip, etc.) result in a forward circular orbiting with an amplitude that increases as a function of load or rotative speed. This mechanism rapidly destroys seals and bearings and often results in full radial rubs. Figure 1 shows the typical forward spectrum cascade of such a machine.

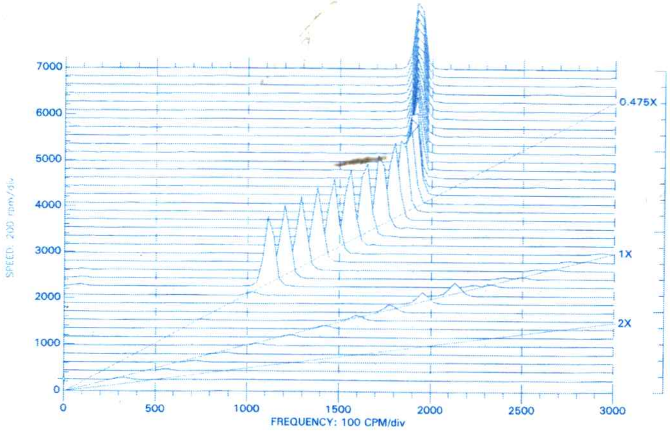


Figure 1
Typical forward spectrum cascade plot of a machine with a fluid-induced instability.

Sometimes, however, the good guys win! Figure 2 shows the forward spectrum cascade of a gas turbine that knows how to fix its own problem very successfully.

As can be seen, a subsynchronous forward circular whip begins at 3900 cpm when rotative speed reaches 8000 rpm. The whip is most severe at rotative speed of 10,000 rpm. Thereafter the forward circular subsynchronous whip gradually drops in amplitude so that it is last seen at 9600 cpm when the rotative speed is at 31,000 rpm. This whip is never again seen in any operating condition of the gas turbine.

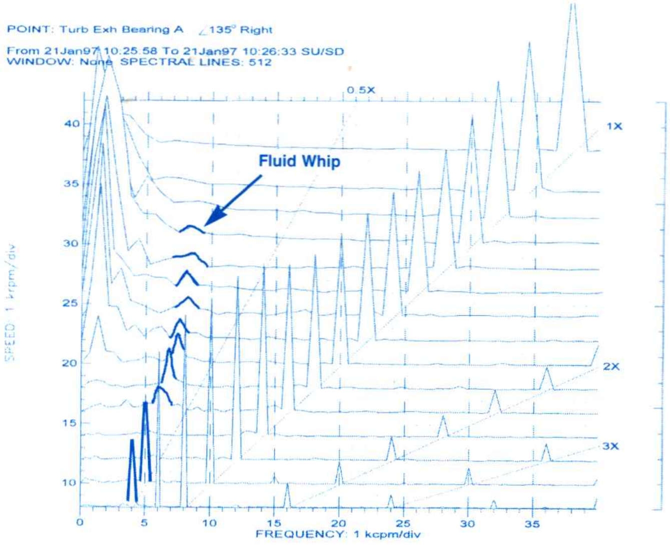


Figure 2
Forward spectrum cascade plot of the gas turbine-generator with fluid-induced whip.

What mechanism causes this whip to cease? I do not know, and since it works well and it is my machine, I really don't care. However, since the mechanism also applies to other machines, we had better know why the system stabilizes! Here are some possibilities.

1. The action at about 1800 cpm, which is the operating speed of the electric generator as well as its pivotal balance resonance speed (with its present support), is so large that the gas turbine shaft is at very high dynamic eccentricity in its bearings. This causes the Direct Dynamic Stiffness to be very high, as well as the fluid swirl ratio (of the sleeve bearing of the gas turbine) to be so low, that the rotor system is stable.

2. This gas turbine has floating sleeve bearings which have not been used on any other machine, to my knowledge, in the last 30 years, and these bearings have some attribute we have not yet noted.

3. The design of the gas turbine is such that a large, steady-state radial sideload is applied to the shaft so that, as the gas turbine comes up to speed, the shaft is pushed to very high eccentricity in the bearing, creating stability as in (1) above. This is reasonably confirmed by the shaft centerline plot of the "A" bearing in Figure 3. Without further information, I believe that this steady radial sideload is the stabilizing element.

I do not know if this gas turbine manufacturer deliberately designed the machine with this stabilizing feature, or if it just happened (like, "Did you play that card, or did it fall out of your hand?"). However, I would bet a lot of money that it was done deliberately, because this manufacturer has an excellent reputation. That's pretty good, because a third of a century ago, when it was built, only about 10% of the present knowledge on the fluid-induced instability was available.■

Remember the ISROMAC Symposium coming up in February 1998 in Hawaii.

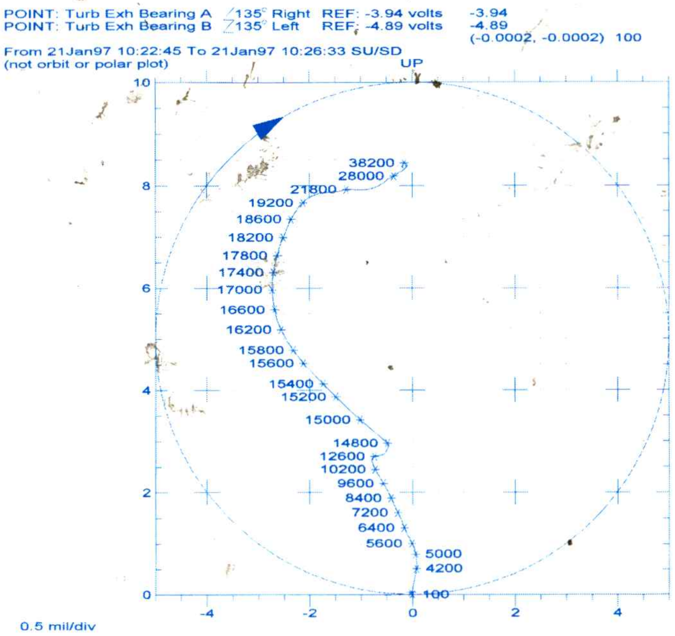


Figure 3
Shaft centerline plot taken during the startup of the gas turbine-generator

Donald E. Bently receives the N.O. Myklestad Award

The Technical Committee on Vibration and Sound (TCVS) of the American Society of Mechanical Engineers (ASME), has selected Donald E. Bently, Chairman and Chief Executive Officer of Bently Nevada Corporation, to receive the organization's prestigious N. O. Myklestad Award. Established in 1991, the award is presented in recognition of major innovative contribution to vibration engineering. TCVS informed Bently that the particular work for which he is being recognized is his original and widely known development of the eddy current probe.

In the garage of his Berkeley, California home in 1955, Bently pioneered an entire industry when he developed the world's first commercially successful eddy current proximity transducer system. Relocating to Minden, Nevada in 1961, his business continued to grow at a rapid rate, soon becoming the State's largest industrial employer. Today, Bently Nevada Corporation products and services are sold worldwide, helping the Company's customers operate more safely and efficiently. Bently Nevada Corporation is the world's leading supplier of monitoring and diagnostic products used by the petrochemical, power generation, pulp and paper, and other industries.

Bently's ongoing research continues to add to the knowledge of rotating machinery behavior. He has published more than 50 papers on the subject of rotating machinery diagnostics through vibration monitoring. Bently received a Bachelor of Science degree in Electrical Engineering with distinction and a Masters degree in Electrical Engineering from the University of Iowa. In 1987, he received an honorary Doctorate in Engineering from the University of Nevada, Reno. He has received numerous other awards and is recognized throughout the world as an authority in his field.

Bently has been invited to personally receive the award at the organization's Vibration Conference to be held in Sacramento, California, 14-17 September.■

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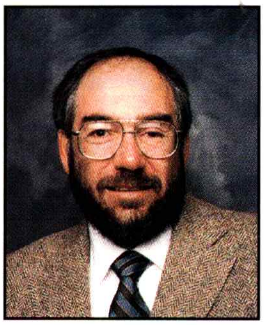
The Synchronous Amplification Factor, Split Resonances, and Virtual Probe Rotation



Charles T. Hatch
Instructor/Developer
Bently Nevada Machinery Diagnostics Technical Training Department

Charlie obtained a B.S. and an M.S. in Mechanical Engineering from the University of California at Berkeley. During his undergraduate education, he worked as an aerospace mechanism designer at Lockheed Missiles and Space Company. His graduate research involved modeling and simulation of an automotive hydraulic valve lifter with oil compressibility effects. After joining Bently Nevada Corporation in 1989, Charlie has worked as an Engineer and Transducer Team Leader in Custom Products, and as a Research Engineer and Laboratory Supervisor in Bently Rotor Dynamics Research Corporation (BRDRC). At BRDRC, Charlie conducted research in the Direct Stiffness of fluid-lubricated bearings, applications of the Root Locus to rotor instability, the influence of fluid-lubricated bearings on polar plot balancing, the behavior of rotor systems with anisotropic properties, and the influence of rotor system gyroscopics on fluid instability. Charlie now works as an Instructor/Developer with the Bently Nevada Machinery Diagnostics Training Department.

The Synchronous Amplification Factor (SAF) is a commonly used measure of how a rotating machine behaves when passing through a balance resonance. It is usually measured using simple, graphical methods on a 1X-filtered Bode or polar plot. However, the observed SAF can become difficult to interpret when split resonances exist in a machine. Also, recent research [1,2] by the Bently Rotor Dynamics Research Corporation (BRDRC) has demonstrated that, when split resonances exist, the measured SAF can become highly dependent on the angular mounting orientation of the observing vibration transducers. Split resonances can also make polar plot balancing more difficult because low speed response vectors may not point toward the heavy spot.

This article discusses a new method that can be used under these conditions to create a set of "virtual" probes at any mounting orientation. These virtual probes can help in evaluating the SAF and also help resolve the polar plot balancing problem.

What is the Synchronous Amplification Factor?

The Synchronous Amplification Factor is measured by the ratio of the vibration amplitude at a balance resonance to the vibration amplitude at speeds well above the resonance. The word "Amplification" refers to the fact that vibration is amplified when a vibrating system passes through a resonance. The term "Synchronous" refers to the fact that the vibration is a result of a rotating force (unbalance) which rotates at the same speed as (is synchronized with) the rotor. A machine with a large SAF will have a sharp, high amplitude balance resonance peak associated with a rapid change in phase. For example, a machine with an SAF of 10 would have a peak amplitude of vibration ten times the vibration level at high speed, well above the resonance. Conversely, a machine with a small SAF will have a relatively low maximum vibration amplitude at resonance, accompanied by a relatively slow change in phase.

The SAF is often considered to be a measure of the damping of a machine, but, actually it is more complicated than that. The maximum amplitude of a rotor system at a balance resonance is controlled by the Quadrature Dynamic Stiffness of the machine. The Quadrature Dynamic Stiffness is a function of both the damping, D, and the Fluid Circumferential Average Velocity Ratio, λ. In machines with a significant amount of fluid circulation, either as process fluid or in fluid-lubricated bearings or seals, the effective damping of the system is almost always significantly smaller than the value of D alone would indicate. Thus, it is more correct to say that the SAF is a measure of the Quadrature Dynamic Stiffness of the rotor system.

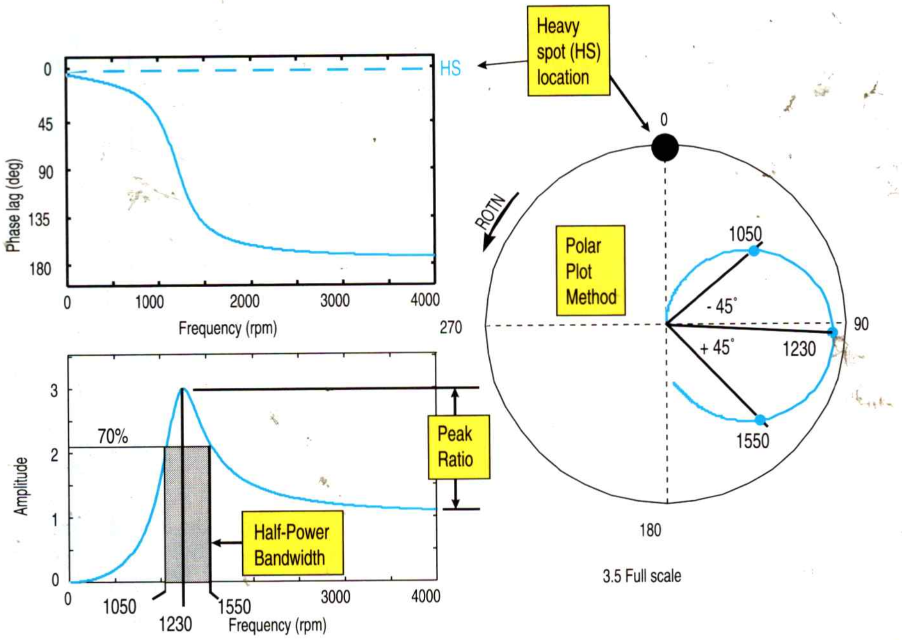


Figure 1
1X-filtered Bode and polar plot of the response of an ideal, isotropic rotor model. The response is as viewed by a vertical probe, and the heavy spot is located at 0° (under the vertical probe) when the Keyphasor pulse occurs. The polar plot from a horizontal probe would appear identical. Three graphical methods for calculating the SAF are shown. The details of the methods are explained in the text.

The SAF has become important in machine acceptance testing. The American Petroleum Institute (API) sets limits on the allowable maximum SAF for various machines. When these limits are incorporated into legal contracts between manufacturers and customers, the measurement of the SAF can become important in satisfying contractual requirements.

There are practical uses for the SAF as well. Because the dominant vibration in most healthy machines is due to unbalance, a high SAF means that a relatively small unbalance can produce large vibration amplitudes in the vicinity of a resonance. Thus, a machine with a high SAF might be more sensitive in the resonance region to changes in balance state for that mode. Also, machines which operate above a balance resonance need to be able to get through the resonance without damage. A machine with a high SAF could be vulnerable to rubbing while passing through a resonance during startup or shutdown. Thus, the SAF can be used as a rough guide for setting alarm limits when running above a resonance.

In the past, the Synchronous Amplification Factor has been considered a measure of the inherent stability of a machine. This is not true. For example, a machine with rolling element bearings might have low Quadrature Dynamic Stiffness and a high SAF. But, if there is no fluid interaction in the machine, the machine would be immune to fluid-induced instability. On the other hand, there are many examples of machines with low SAFs which are plagued with fluid instability problems.

Measurement of the Synchronous Amplification Factor

There are three graphical methods which can be used to measure the SAF. Two of the methods use the amplitude part of a 1X-filtered Bode plot, and one method uses a 1X-filtered polar plot (Figure 1) [3].

The Half-Power Bandwidth method is endorsed by the API. The term "Half-Power" originated in AC electrical circuit theory, where the -3dB point corresponds to a voltage oscillation amplitude of 0.707 times the amplitude at resonance. Since simple electrical and mechanical oscillators are mathematically equivalent, the method can be applied to rotor systems; thus, the half-power points of the Bode plot have a vibration amplitude of about 70% of the maximum balance resonance amplitude. Once these two points have been identified, they are used to find the corresponding speed bandwidth. The difference between the half-power rotor speeds on each side of the resonance (Ω_{high} and Ω_{low}) defines the half-power bandwidth. These two rotor speeds are used together with the speed of maximum amplitude (the resonance speed Ω_{res}) to calculate the SAF:

$$SAF = \frac{\Omega_{res}}{\left(\Omega_{high} - \Omega_{low}\right)} \tag{1}$$

The Peak Ratio method also uses the amplitude part of a 1X-filtered Bode plot. For an ideal rotor system, the ratio of the peak vibration amplitude at the balance resonance to the vibration amplitude at infinite running speed is equal to the SAF. In practice, the SAF can be measured by taking the peak vibration amplitude (A_{peak}) and dividing it by the vibration amplitude at a speed well above resonance (A_{fast}):

$$SAF = \frac{A_{peak}}{A_{fast}} \tag{2}$$

Finally, a third method can be used, based on a 1X-filtered polar plot. For an ideal rotor system, the half-power points have a phase of 45° ahead of and 45° behind the phase at resonance. To use this method, a line is drawn from the origin of a compensated polar plot to the maximum amplitude of the resonance. Then, two additional lines are drawn from the origin at 45° left and 45° right of the first line. The rotor speeds associated with these three points are used in the same way as for the half-power bandwidth method:

$$SAF = \frac{\Omega_{res}}{\left(\Omega_{+45^\circ} - \Omega_{-45^\circ}\right)} \tag{3}$$

One good thing about these three methods is that the graphical measurements are independent of each other. Thus, they can be used as a check against mistakes in the calculations. Another thing to remember is that, because all real rotor systems deviate to some extent from the ideal, all three of these methods are approximate.

Several things can influence the accuracy of these measurements: uncompensated data, closely spaced shaft and casing modes, and split resonances. All plots must be compensated for slow roll runout before attempting the calculations. Closely spaced modes especially affect the Peak Ratio method, where the "high speed" response may be covered up by a higher mode or a mode from an adjacent machine. Split resonances can make the measurement of the SAF both difficult and dependent on angular orientation of the transducer used to measure the vibration.

Split resonances and the Synchronous Amplification Factor

A split resonance consists of two close resonances with a similar mode shape which are separated in speed. This is in contrast to two resonances which are independent modes and may have significantly different mode shapes.

Split resonances occur when a rotor system has anisotropic (asymmetric) stiffness; in other words, the rotor system static direct stiffness is not the same in all radial directions.

The rotor system direct stiffness consists of a combination of a large number of components which are chained together. It includes everything between "ground" and the rotor mass: foundation stiffness, bearing pedestal stiffness, the stiffness associated with any externally attached piping system, fluid bearing oil wedge stiffness, seal stiffness, and any rotor stiffness.

All of these components can have stiffness characteristics that are different in different radial directions. For example, bearing support pedestals on horizontal machines are often weaker horizontally than vertically. External

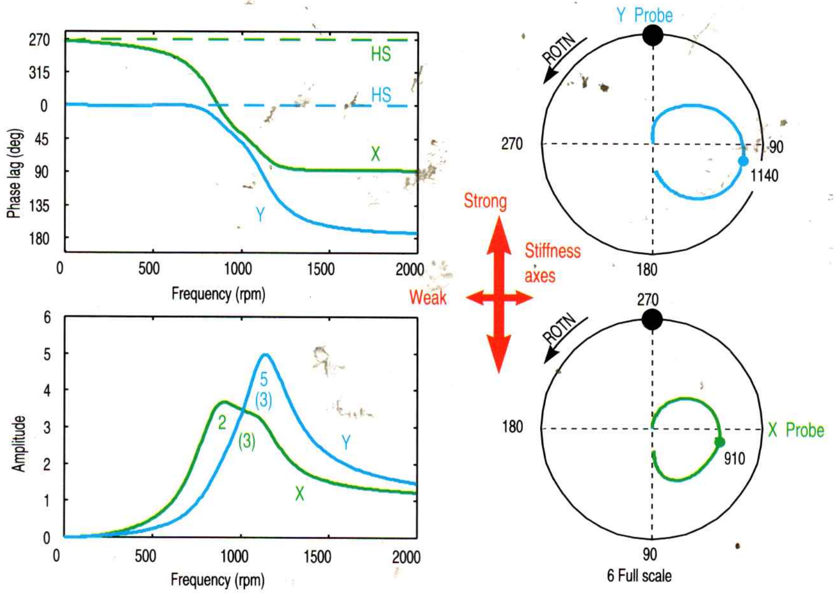


Figure 2

1X-filtered Bode and polar plots of a rotor system modeled with anisotropic stiffness. The measured responses from both coplanar probes are shown. The stiffness is weak horizontally and strong vertically. The probes are aligned with the stiffness axes. Note the difference in appearance of the two polar plots and that the split resonance peaks occur at different speeds. The numbers under the resonance peaks indicate the observed SAF using the Half-Power Bandwidth method and the Peak Ratio method (parentheses).

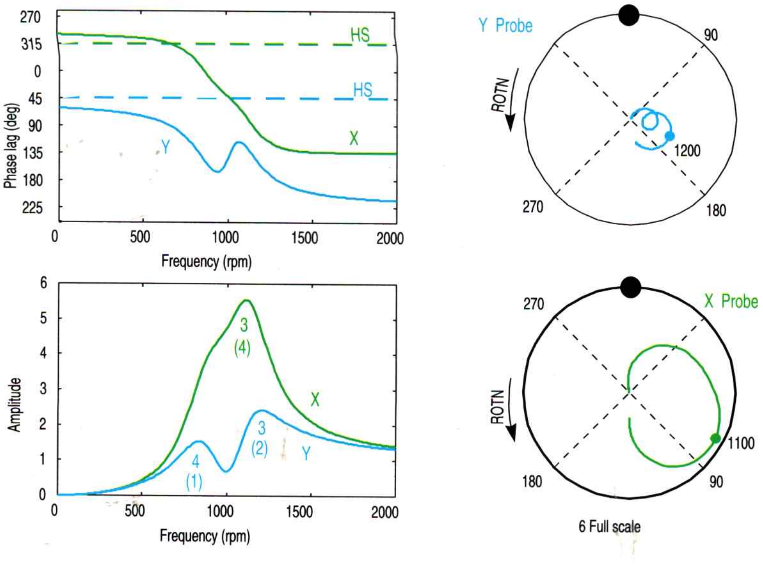


Figure 3

1X-filtered Bode and polar plots from the same model as in Figure 2. The probes have been rotated +45° (X to Y) relative to the horizontal and vertical stiffness axes. Note the internal loop in the Y-probe data which suggests a structural resonance. This loop is purely a result of the split resonance and the chosen viewpoint. Also note the large difference in the appearance of the resonance peaks. The numbers under the peaks indicate the observed SAF using the Half-Power Bandwidth method and the Peak Ratio method (parentheses). Note that the observed SAFs are different than those shown in Figure 2.

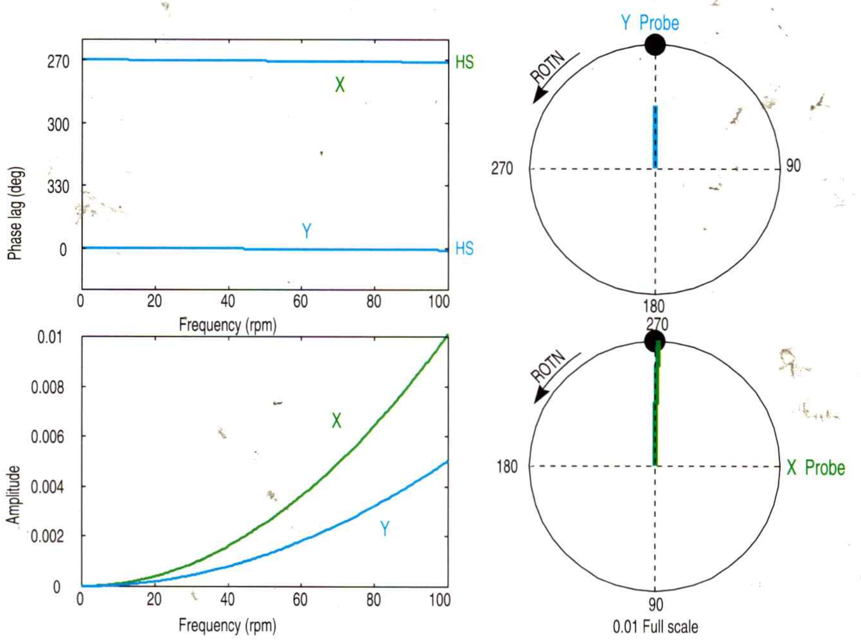


Figure 5
The low speed response of the anisotropic rotor model shown in Figure 2. The probes are now aligned with the stiffness axes, and the low speed phase lag angles point toward the heavy spot.

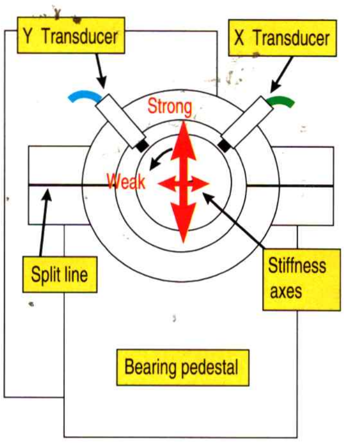


Figure 6
Schematic of a horizontally split machine. The tall bearing pedestal creates a rotor system support which is weaker horizontally than vertically. Probes are mounted ±45° from the vertical. This situation results in misalignment between the probe mounting orientation and the anisotropic stiffness axes, a situation modeled in Figure 3.

very difficult to interpret. This has important implications for machine acceptance testing.

Anisotropy can also lead to difficulty when trying to identify the location of the heavy spot for balancing. Figure 4 shows the low speed part of a polar plot from an anisotropic rotor model with the probes rotated 45° from the stiffness axes. The low speed phase does *not* point toward the heavy spot; both the X and Y probe plots have a phase distortion related to the degree of anisotropy and the probe orientation. Figure 5 shows the low speed behavior for the same model when the probes are aligned with the stiffness axes. In this case, the probes are mounted horizontally and vertically to coincide with the horizontal and vertical stiffness axis orientation. Note that the low speed phase from both probes now points towards the heavy spot. Analysis [1] has shown that, for synchronous behavior, ***when a pair of XY probes are aligned with the anisotropic stiffness axes, the low speed phase will be in the direction of the heavy spot.*** Reference [3] describes how the presence of a nearby fluid-lubricated bearing can also affect polar plot balancing.

Some machines, such as horizontally split compressors, are mounted on vertical bearing support pedestals. To avoid the split line in this type of machine, XY probe pairs are usually mounted with an angular orientation ±45° from the vertical (Figure 6). Because of the vertical bearing support pedestal structure and the gravity load of the rotor supported on fluid-lubricated bearings, the orientation of the stiffness axes may be close to horizontal and vertical (this assumes no other significant influences, such as process loads or misalignment). This is the same situation that was modeled in Figure 3. Because the probe axes are rotated relative to the rotor system stiffness axes, we would expect to see significant differences between the data from the X and Y probes in such a machine.

Virtual Probe Rotation

Given the sensitivity of vibration data to anisotropic stiffness and probe orientation, probes should be mounted so they are aligned with the stiffness axes. However, even if it were physically possible to do this, changes in the machine load, alignment, or support structure could cause the orientation of the stiffness axes to change.

Fortunately, the vector data from any XY probe pair can be mathematically transformed [1] to create a set of "virtual" probes at any desired angular orientation. Virtual Probe Rotation has been incorporated into the latest releases of Bently Nevada ADRE® and Data Manager® 2000 Software. Virtual Probe Rotation allows the user to examine the rotor response from many different probe mounting orientations to get a clearer picture of what is happening as the machine passes through resonances. Because measurement of the SAF can be very dependent on probe orientation, the user can select an orientation which provides the most appropriate viewpoint for calculating the SAF.

Virtual Probe Rotation can also be very helpful in identifying the location of the heavy spot for balancing machines with anisotropic stiffness. Probes can be mathematically rotated until the low

> ***"Virtual Probe Rotation, now available in Bently Nevada ADRE® and Data Manager® 2000 Software, offers a new, powerful analysis tool for the machinery manager and diagnostician."***

speed response vectors of the compensated X and Y polar plots point in the same direction. That direction will be the direction of the heavy spot. When the probes are aligned with the stiffness axes, the compensated low speed phases of the probe pair on compensated Bode plots will differ by approximately 90°. (When the probes are *misaligned* with stiffness axes, the low speed phase can differ by considerably more or less than 90°.)

It must be emphasized that Virtual Probe Rotation requires vector data from two orthogonal, coplanar probes. Also, the two transducers must be of the same type (such as displacement). Data from single probes cannot be transformed in this manner.

This brings up a good point. The figures presented here show that a one-probe-per-plane machine monitoring system, while better than nothing, is quite limited in its ability to show what is happening in rotor systems with a significant degree of anisotropic stiffness. As we have seen, the polar and Bode plots from two orthogonal probes can appear quite different from each other. A diagnostician who used only the Y probe data from Figure 3 would have a very misleading picture of what was actually happening in the machine.

Summary

There are three graphical methods for estimating the Synchronous Amplification Factor of a machine. The methods are independent, but all three are derived from ideal rotor models. Because real rotor systems deviate from ideal behavior, all of these methods are approximate.

Anisotropic rotor system stiffness tends to produce elliptical orbits and split resonances. When split resonances are present, Bode and polar plots from X and Y probes in the same plane can appear very different. Also, the appearance of Bode and polar plots will be different, depending on the relative angular orientation of the probes and stiffness axes. The combination of these effects can make determination of the SAF difficult and viewpoint dependent. This has important implications for machine acceptance testing.

Rotor system support anisotropy, because it affects measurement of low speed phase, can also make polar plot balancing more difficult. In anisotropic rotor systems, the low speed response vectors of X and Y polar plots may not agree with each other as to the location of the heavy spot.

Vector data from a coplanar XY probe pair can be mathematically transformed to create a set of rotated, "virtual" probes at any desired angular orientation. Virtual Probe Rotation allows us to "view" the rotor from several different aspects and observe changes in the observed SAF.

Virtual Probe Rotation is also useful for polar plot balancing of anisotropic machines. By rotating the virtual probes to align them with the stiffness axes, the low speed phases of a pair of XY probes will agree as to the location of the heavy spot. Thus, it becomes easier to make a decision on initial weight placement in the balancing process.

Virtual Probe Rotation, now available in Bently Nevada ADRE and Data Manager 2000 Software, offers a new, powerful analysis tool for the machinery manager and diagnostician. ■

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Some realities of field balancing

by Robert C. Eisenmann, P.E.
President
Machinery Diagnostics, Inc.
Minden, Nevada

While many turbomachinery rotors are successfully shop-balanced at low or high speed, some rotors can only be balanced in their casings in the field, using every technique, trick, and piece of information available to the specialist.

Most turbomachinery rotors are successfully balanced in slow speed shop balancing machines. This approach provides good access to all correction planes, and the option of performing multiple runs to achieve a satisfactory balance. It is generally understood that running at slow speeds with simple bearings or rollers does not duplicate the rotational dynamics of the field installation. Other rotors are shop balanced on high speed machines installed in vacuum chambers. These machines provide an improved simulation of the installed rotor behavior, due to the higher speeds and the use of bearings that resemble the normal running bearings. In these high speed chambers, the influence of blade and wheel aerodynamics is significantly reduced by operating within a vacuum.

Some machines, such as large steam turbines, often require a field trim balance, due to the influence of higher order modes or the limited sensitivity of the low speed balancing techniques.

There is also a small group of machines, which, except for component balancing, must be balanced in the field. These machines have segmented rotors, that are assembled concurrently with the diaphragms and/or casing. In these machines, the final rotor assembly is not achieved until most of the stationary elements are bolted into place. An example of this type of rotor is shown in Figure 1.

Machine configuration

This rotor consists of an overhung hot gas expander wheel, a pair of midspan compressor wheels, and three stages of overhung steam turbine wheels. A series of axial through bolts are used to connect the expander stub shaft through the compressor wheels and into the turbine stub shaft. This type of rotor assembly is similar to many gas turbine rotors. However, in this machine, the rotor must be built concurrently with the inner casing. Specifically, the horizontally split internal bundle is assembled with the titanium-aluminum compressor wheels, stub shafts, plus bearings and seals. End casings are attached, the expander wheel is bolted into position, and the three turbine stages are attached with another set of through bolts.

The eight rotor segments are joined with Curvic® couplings, which are identified as #1 through #7 on Figure 1. Even with properly ground and tight fitting

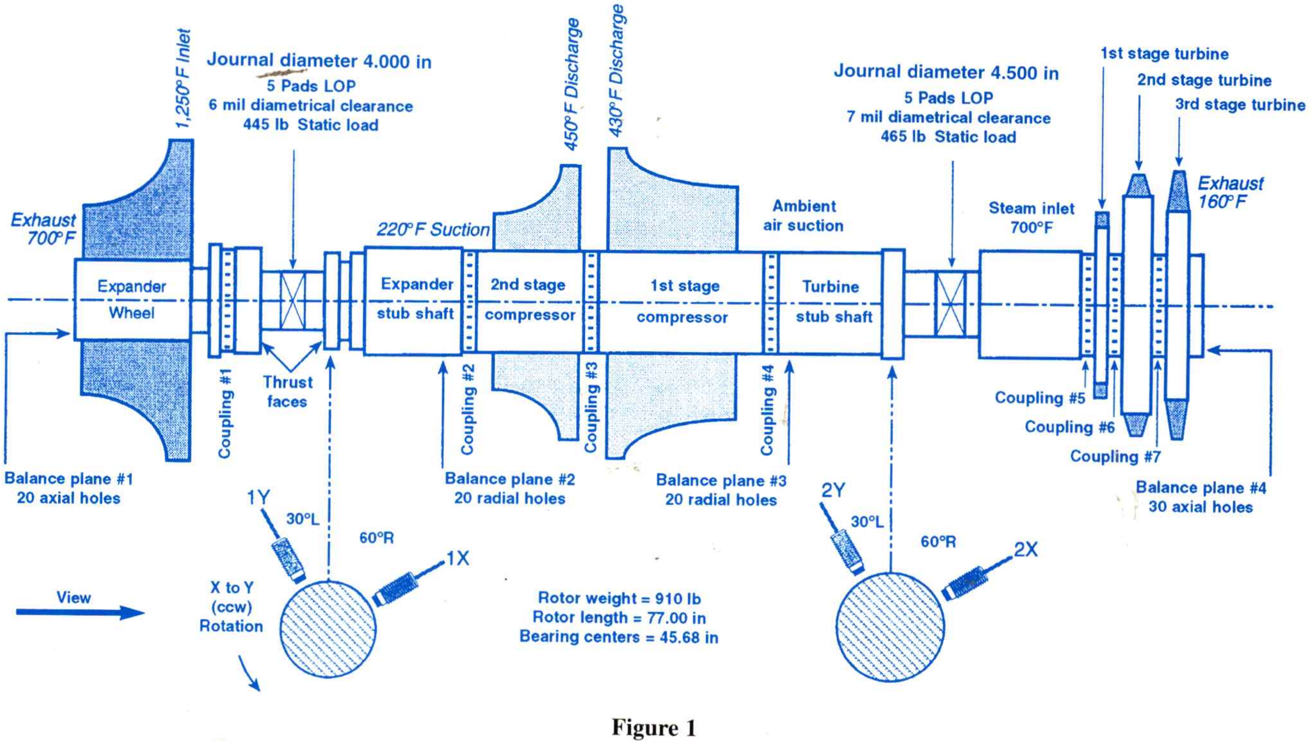

1,250°F Inlet
Journal diameter 4.000 in
5 Pads LOP
6 mil diametrical clearance
445 lb Static load
450°F Discharge
430°F Discharge
Journal diameter 4.500 in
5 Pads LOP
7 mil diametrical clearance
465 lb Static load
1st stage turbine
2nd stage turbine
3rd stage turbine
Exhaust 700°F
220°F Suction
Ambient air suction
Steam inlet 700°F
Exhaust 160°F
Expander Wheel
Expander stub shaft
2nd stage compressor
1st stage compressor
Turbine stub shaft
Coupling #1
Thrust faces
Coupling #2
Coupling #3
Coupling #4
Coupling #5
Coupling #6
Coupling #7
Balance plane #1
20 axial holes
Balance plane #2
20 radial holes
Balance plane #3
20 radial holes
Balance plane #4
30 axial holes
1Y
30°L
60°R
1X
2Y
30°L
60°R
2X
View
X to Y
(ccw)
Rotation
Rotor weight = 910 lb
Rotor length = 77.00 in
Bearing centers = 45.68 in

Figure 1
Expander-air compressor-steam turbine rotor and vibration transducer arrangement.

couplings, there is potential for relative movement of rotor elements. Although each of the rotor segments is component balanced, any minor shift between elements will produce a synchronous unbalance force. Since this unit operates at 18,500 rpm, a few grams of unbalance or a mil or two of eccentricity will result in excessive shaft vibration and a strong potential for machine damage.

Furthermore, the distribution of operating temperatures noted on Figure 1 reveals the complexity of the thermal effects that must be tolerated by this unit. The expander wheel inlet temperature is 677°C (1,250°F), and compressor discharge temperature is in excess of 221°C (430°F). The steam turbine operates with a 371°C (700°F) inlet temperature and a final exhaust temperature of 71°C (160°F).

By any definition, this must be considered a difficult rotor to deal with. More than one person has referred to this rotor as a "squirrel," meaning a rotor which responds erratically. On the positive side, this machine is a compact design that yields a high thermal efficiency. Therefore, when the unit is properly assembled, and balanced, it is very cost-effective to operate.

Field balancing this 413 kg (910 lb) rotor is achieved by adding or removing weights from a series of threaded holes in four balance planes (Figure 1). Balance Plane #1 contains 20 axial holes at the expander exhaust. Plane #2 and #3 each have 20 radial holes. They are located in two rows on the expander and turbine stub shafts, respectively. The outboard end of the turbine 3rd Stage wheel contains balance Plane #4 with 30 axial holes.

Shaft vibration measurements are made with Bently Nevada XY proximity probes (Figure 1). These radial probes are connected to a Bently Nevada 3300 Monitoring System, which in turn is connected to a Transient Data Manager® (TDM2) Communications Processor (CP). The CP sends static and dynamic signals from the monitors to a personal computer running TDM2 Software.

This instrumentation is used for monitoring, trending, and machinery diagnostics. The 3300 Monitors can automatically activate alarms when vibration setpoints are exceeded, and the computer system has additional Alert and Danger indications. This instrumentation also provides the steady state and transient vector data necessary for field trim balancing this complex rotor.

Balancing calculations

During shop balancing, it is easy to make a weight change; it is inexpensive to make a run, and there is little physical risk to the machinery or the operator. In the case of field balancing, it is often difficult to change weights, and it is generally expensive to make a full speed run. Furthermore, if an incorrect weight is used, the results may be hazardous to the machinery, and the health of the operator.

Each field balance shot should be a meaningful move, and should contribute to the overall database describing the behavior of the machine. Balancing behavior is based on the fundamental relationship between the shaft response and the applied force expressed by:

$$\textit{Response} = \frac{\textit{Force}}{\textit{Restraint}} = \frac{\textit{Force}}{\textit{Dynamic Stiffness}} \tag{1}$$

This general expression has many specific applications in rotor dynamics. Within the balancing discipline, the **response** is the measured shaft or casing **motion** and the **force** is the magnitude of the unbalance force. The **restraint** is the dynamic stiffness. Another way to view this term is to consider it as the sensitivity of the machine to rotor unbalance.

The variables in Eq.1 are vector quantities. That is, each parameter carries both a magnitude and a direction. If the vibration of a rotor is described by the A vector, with amplitude units of mil pp, and the unbalance magnitude is defined by the U vector with units of grams instead of force, then the sensitivity S vector carries the anticipated engineering units of gram/mil pp. Equation 1 may now be rewritten as follows:

$$A = \frac{U}{S} \tag{2}$$

The vibration vector A is measured directly, and the problem resolves to one of determining the unbalance based upon an unknown Sensitivity vector. According to most sources, this problem was originally solved by E.L. Thearle in his 1934 ASME paper. He basically provided a way to practically determine the sensitivity vector. That procedure involves the addition of a known calibration weight to the rotor and determining the response vector C due to that calibration weight. If the calibration weight vector is defined by W, and the resultant vibration response vector of the rotor is defined as the $A+C$, Equation 2 may be expanded as follows:

$$A + C = \frac{U+W}{S} = \frac{U}{S} + \frac{W}{S} = A + \frac{W}{S}$$

or, by subtracting A from both sides,

$$C = \frac{W}{S}$$

This last expression may be solved for the sensitivity S vector:

$$S = \frac{W}{C} \tag{3}$$

The S vector may now be combined with the initial vibration response vector in Eq. 2, and the unbalance vector computed. The concept has been around for over 60 years, and has been successfully applied on many machinery rotors.

The previous discussion covered only one measurement plane. The same logic may be applied to two or more measurement and correction planes. With respect to the current rotor under investigation, there are only two measurement planes, and four separate correction planes. This is different from most machines where the number of measurement and correction planes are equal.

Thus, the general form of eq 3 is:

$$S_{mp} = \frac{W_p}{C_{mp}} \tag{4}$$

where m is the measurement plane and p is the weight correction plane.

The balancing technique of *"add a weight, make a run, take data, and do vector calculations"* generally produces acceptable results on many machines. However, this technique may fail on a complex rotor. A review of the balance history of the machine may help identify the correct technique.

Historical data review

A double overhung rotor with an appreciable midspan mass (such as Figure 1) has the potential for multiple resonances with both forward and reverse modes. In order to better understand the behavior of this machine, various historical data sets were reviewed. Fortunately, TDM2 data was available. This included steady state orbits, plus transient speed Bode and polar plots. From this data it was clear that reverse orbits appeared around 7,000 and 17,000 rpm. In addition, tabular lists of sampled vectors were also documented. **It was noted that the slow roll vectors were not repetitive, and this was a cause for concern.**

Previous field balancing activities on this machine were generally successful when a two step correction was used. The first step consisted of a balance at the process hold point of 14,500 rpm, using the outboard Planes #1 and #4. This was followed by a trim at 18,500 rpm on the inboard Planes #2 and #3 located next to the compressor wheels. It was evident that if the rotor wasn't adequately balanced at 14,500 rpm, it probably wouldn't run at 18,500 rpm.

For comparative purposes, the Sensitivity vectors from three different data runs were computed based upon steady state data at 14,500 rpm. These $\boldsymbol{S}$ vectors were computed in accordance with Eq. 4. The results are summarized in Table 1. The two shaded vectors in this table are of questionable accuracy, due to the fact that the differential vibration vector $\boldsymbol{C}$ was less than 0.1 mil pp. This small differential vibration is indicative of low sensitivity to the applied weight.

The $\boldsymbol{S}$ vectors in Table 1 have some similarities, but the variations are significant. For example, the magnitude of $\boldsymbol{S}_{12}$ varies from 22.6 to 76.4 gram/mil, and a 76° angular difference is noted. On $\boldsymbol{S}_{24}$, the amplitudes change from 24.2 to 14.9 gram/mil, but the angles reveal a 59° spread. **At this point, a preliminary conclusion might be reached that this rotor is nonlinear, and cannot be field balanced.** Clearly, additional information was needed to define the rotor behavior.

Process observations

Further examination revealed that vibration severity changed in accordance with the machinery operational state. Peak vibration amplitudes occur at a rotor critical (balance resonance) that appears between 7,600 and 8,100 rpm. This resonance displays the following characteristics versus operating conditions:

- Cold startup to 14,500 rpm
  Peak response of 2.0 to 5.0 mil pp
- Warm coastdown from 14,500 rpm
  Peak response of 4.0 to 5.0 mil pp
- Hot shutdown from 18,500 rpm
  Peak response of 6.0 to 8.0 mil pp

These amplitude variations, combined with changes in the rotor Synchronous Amplification Factor through the resonance, suggest a change in the damping term (Quadrature Dynamic Stiffness) of dynamic stiffness.

Furthermore, it was determined that the different operating conditions were directly associated with variations in shaft slow roll vectors, which were taken at 1,000 rpm. In all cases, the runout vectors for each cold startup, warm coastdown or hot emergency shutdown were consistent within each condition. When the rotor was shut down from a hot state, the slow roll vectors always returned to the same initial starting vectors as the rotor cooled. **Therefore, the apparent changes in slow roll were resolved.**

It was also determined that the previous vibration response vectors used for balancing were acquired at a process hold point of 14,500 rpm. Under this condition, the machine speed was held constant, but rotor and casing temperatures were changing as the process stabilized prior to continued startup of the plant. Next, the complete transient response picture needed to be added to this process data.

Transient data review

There is an adage that states, *"If your only tool is a hammer, then all of your problems begin to resemble nails."* This is very appropriate for this particular rotor. If the *"put a weight in and take a weight out"* balance technique is the only tool available, the results are often disappointing. This is evident from the spread in sensitivity vectors shown in Table 1. Attempting to balance a machine when these values keep changing is difficult at best. Many runs are required to attain a barely acceptable balance state.

It is quite clear that additional tools must be employed to properly field balance this machine. Certainly, the plant process observations discussed in the

S Vector	Data Set #1 (gram/mil @ deg)	Data Set #2 (gram/mil @ deg)	Data Set #3 (gram/mil @ deg)
S_{11}	20.3 @ 139°	16.2 @ 179°	*Not Available*
S_{12}	48.1 @ 34°	22.6 @ 309°	76.4 @ 233°
S_{13}	42.6 @ 211°	14.7 @ 177°	24.6 @ 200°
S_{14}	34.1 @ 259°	41.7 @ 289°	16.2 @ 305°
S_{21}	18.3 @ 308°	13.4 @ 345°	*Not Available*
S_{22}	19.1 @ 168°	18.5 @ 142°	18.6 @ 160°
S_{23}	32.3 @ 258°	20.2 @ 269°	31.2 @ 221°
S_{24}	24.2 @ 83°	20.3 @ 147°	14.9 @ 142°

Shaded vectors are of questionable accuracy due to small differential vibration vectors with calibration weights installed

Table 1
Sensitivity vectors based on steady state data at 14,500 rpm.

previous section are significant. However, this information must be supplemented by an examination of the variable speed (transient) vibration response data (Bode and polar plots), plus an understanding of the rotor critical (balance resonance) speeds, and mode shapes.

The vibration response from the installed proximity probes was extracted from the TDM2 database; a typical startup Bode plot is shown in Figure 2. This data displays the Y axis probes from both measurement planes. Both plots are corrected for slow roll runout. (Note: the Bode plots start at 4,000 rpm for better resolution.) Thus, the resultant data is representative of the true dynamic motion of the shaft at each of the two lateral measurement planes.

The major resonance occurs at approximately 7,800 rpm. Note that the process hold point at 14,500 rpm displays substantial amplitude and phase excursions. This is logically due to the heating of the rotor and casing, plus variations in "settle out" of the operating system (i.e., pressures, temperatures, flow rates, and molecular weights). **Although this process stabilization is a necessary part of the startup, the variations in vibration vectors negate the validity of this information for use as repetitive balance response data.**

At speeds above 14,500 rpm, there are additional vector changes, and a desirable *flat spot* (plateau) in the amplitude and phase curves does not appear. The only essentially flat area that provides consistent vibration vectors occurs in the vicinity of 14,000 rpm.

To test the validity of this conclusion, individual vectors at 14,000 rpm were extracted from the TDM2 transient startup files. These vibration vectors, in conjunction with the installed weights, were used to compute the sensitivity vectors in accordance with Eq. 4. The results of these computations are presented in Table 2. These vectors are directly comparable to the vectors discussed in Table 1. However, it is clear that the consistency of the ***S*** vectors among the three data sets is far superior in the results shown in Table 2. **Therefore, future balance corrections at this intermediate point were based upon the transient vectors acquired by the TDM2 System as the rotor passed through 14,000 rpm.**

The Bode plot in Figure 2 also exhibits a variety of amplitude and phase changes between 14,500 and 18,500 rpm. Other vector changes are visible as the machine approaches the normal operating speed of 18,500 rpm. Some of these changes are due to the influence of a backward mode around 17,000 rpm.

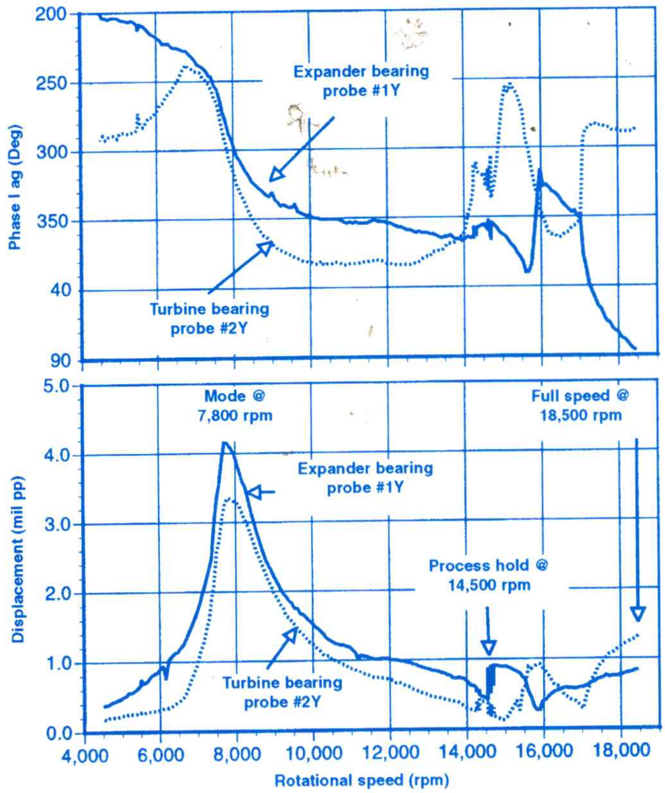


Figure 2
Runout-compensated Bode plot of Y-axis probes during typical machine train startup.

S **Vector**	**Data Set #1** (gram/mil @ deg)	**Data Set #2** (gram/mil @ deg)	**Data Set #3** (gram/mil @ deg)
S_{11}	24.7 @ 144°	86.2 @ 188°	*Not Available*
S_{12}	125. @ 333°	38.2 @ 228°	42.4 @ 228°
S_{13}	31.8 @ 207°	20.8 @ 184°	22.9 @ 176°
S_{14}	21.8 @ 322°	36.1 @ 321°	30.7 @ 301°
S_{21}	24.9 @ 313°	21.6 @ 311°	*Not Available*
S_{22}	17.3 @ 167°	23.7 @ 175°	18.9 @ 178°
S_{23}	78.7 @ 299°	22.2 @ 221°	27.4 @ 232°
S_{24}	27.6 @ 147°	25.8 @ 166°	28.3 @ 164°

Shaded vectors are of questionable accuracy due to small differential vibration vectors with calibration weights installed

Table 2
Sensitivity vectors based on transient data at 14,000 rpm.

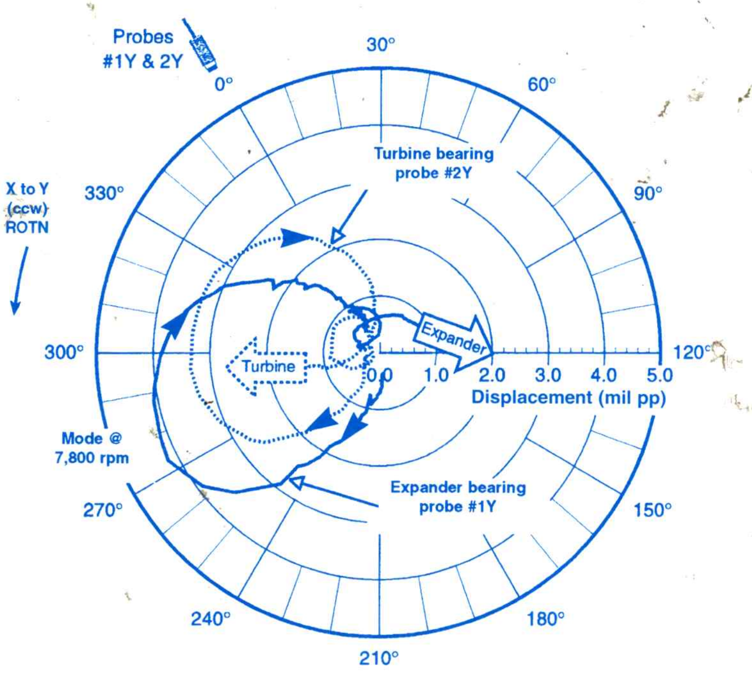


Figure 3
Runout-compensated polar plots of Y-axis probes during typical machine startup.

This higher speed data is difficult to fully comprehend in the Bode plot. It becomes more definitive when this same data is plotted in a polar format (Figure 3).

Both Y axis probes show the primary rotor resonance at 7,800 rpm. The point of major interest is that, at full speed, the turbine-end shaft is moving towards the 9 o'clock direction, and the expander-end shaft is heading towards 4 o'clock. This behavior indicates the presence of a pivotal mode occurring at a frequency above 16,000 rpm.

In many cases, this type of response would not be unusual. However, for this unit, the historical files had no indication of a resonance around the normal running speed. Due to the response of midspan balance weights (Planes #2 and #3) at 18,500 rpm, it was clear that the vibration data was correct, and the historical undamped mode shapes were flawed.

Analytical model

As was previously stated, there are only two lateral measurement planes along the entire 2 metre (79 inch) length of this rotor. Since there were no other feasible locations for shaft probes, additional measurement options were eliminated. Therefore, the only viable approach resided with a meaningful analytical model of this rotor system.

A 65-station damped model was constructed. This computer model included bearing stiffness and damping that varied with speed, plus flexible bearing supports. The calculations produced eigenvalues (natural frequencies) plus eigenvectors (mode shapes).

The first four modes are stiff shaft pivotal and translational shapes. These modes did not appear in the vibration data due to the high dynamic stiffness for each mode. A backward mode was detected at 7,080 rpm. This mode was not visible in the startup plots, but it briefly appears as a reverse precession orbit in some of the hot coastdown data. The most active forward mode within the operating speed range occurs at a damped frequency of 7,838 rpm.

The calculated mode shapes for this resonance are shown in Figure 4. Note that the normalized deflection at both bearings are quite small. This indicates minimal motion of the journals within their respective bearings. With small relative motion, the velocity is low, and the effect of bearing damping is minimal.

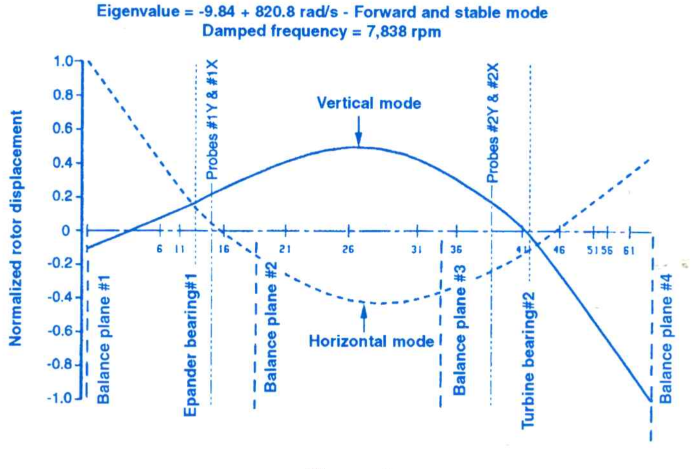


Figure 4
Calculated damped mode shapes at main rotor resonance at nominally 7,800 rpm.

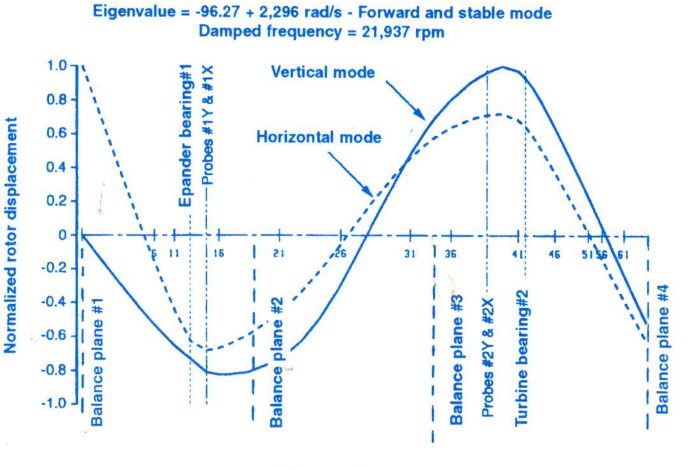


Figure 5
Calculated damped mode shapes at rotor resonance occurring above operating speed.

The validity of this analytical model is supported by the correlation of the computed resonance frequency of 7,838 rpm (Figure 4), with the measured resonance of 7,800 rpm seen in the Bode plot (Figure 2). It is also clear from Figure 4, that the rotor balance response at this resonance can be effectively controlled by corrections at the modally effective end Planes #1 and #4.

The analytical model revealed a backward mode at approximately 17,700 rpm. This pivotal mode is visible by a phase reversal (in phase to out of phase) on the transient data (Figures 2 and 3). Immediately above the normal operating speed of 18,500 rpm, a damped mode was calculated at a frequency of 21,937 rpm (Figure 5). This mode has the same deflection characteristics as the mode at 17,700 rpm. In both cases, the inboard balance Planes #2 and #3 are the most modally effective correction planes for this speed domain.

It was also concluded that weight corrections adjacent to these compressor wheels (Planes #2 and #3) should be out of phase. This was because a nodal point exists at the middle of the rotor (Figure 5) between the planes. The validity of this conclusion was field tested on the machine. The installation of a pair of weights at the middle planes at the same angle resulted in excessive vibration. However, a couple shot (out of phase) proved to be smooth, and supportive of the analytical mode shape at high speed.

Finally, the existence of a resonance at slightly above running speed, was previously noted on the polar plot (Figure 3). The analytical results shown in Figure 5 support this earlier observation. Once again, vibration measurements and the analytical tools are combined to explain the behavior of a complex machine.

Overall balance improvement

Armed with an improved understanding of the shaft response and mode shapes, plus a better appreciation of the process influence, the rotor was balanced at the intermediate speed of 14,000 rpm by using TDM2 transient data acquired during cold startups. This intermediate step was a two plane balance with weight corrections at the outboard Planes #1 and #4. This allowed the rotor to run at full speed of 18,500 rpm. A final two plane trim balance was performed on the interior Planes #2 and #3 after a full heat soak.

Transducer	Before balancing @ 14,025 rpm	After balancing @ 18,500 rpm
Expander bearing #1Y	1.41 mil pp @ 139°	0.31 mil pp @ 173°
Expander bearing #1X	1.10 mil pp @ 34°	0.50 mil pp @ 29°
Turbine bearing #2Y	0.80 mil pp @ 298°	0.11 mil pp @ 25°
Turbine bearing #2X	0.79 mil pp @ 199°	0.37 mil pp @ 192°

Table 3
Comparison of initial low speed versus final high speed runout-compensated vibration vectors.

The initial shaft vibration vectors at 14,025 rpm, and the final vectors at 18,500 rpm are shown in Table 3. The magnitude of the final running speed vectors ranged from 5 to 8% of the diametrical bearing clearance. The suitability of this balance state is demonstrated by an extended process run on this machine.

Conclusions

From the previous discussion, and the machinery example, it is clear there are some definitive guidelines that should always be considered on complex rotors. Specifically, the following items are summarized for future reference:

- Slow roll vectors, acquired under the same conditions, must always repeat. If the slow roll vectors change from run to run, then other influences or problems (e.g. a cracked shaft) should be considered.
- Transient startup and coastdown data must always be obtained. Bode and polar plots should be used to examine the 1X synchronous response for resonance and mode shape information. Spectrum cascade plots should be reviewed to verify that other frequencies are nonexistent, or are not generated as a result of the balance work.
- Balance speeds should be selected in relative plateau regions that are removed from regions that exhibit amplitude and phase variations.
- Calculations should be made for both Y axis and X axis shaft probes. When flexible supports are encountered, calculations should also be performed for the casing vibration transducers.
- Weight correction locations should be modally effective. If a weight change produces an unexpected result, the presumed mode shape may be incorrect.
- When adding (or removing) calibration weights, check the response vector ***C***. Small vector changes indicate a lack of sensitivity to the weight.
- Sensitivity vectors, acquired under similar conditions, should be consistent. If significant changes are noted, look for problems in the system.
- Care should be taken when analytically modeling the rotor system. Small mistakes or improper assumptions can result in major errors in mode shapes, and frequencies.
- Some machines, as demonstrated in this example, will require an intermediate speed, and a final trim balance speed.
- Do careful work, and be rewarded by the satisfaction of excellent results, plus the ability to "*balance a squirrel.*" ■

(Additional text in sidebar on page 19).

Machinery Messages

Proper site management important in today's competitive marketplace

by Brent Tessmann
Product Service Manager, Americas
Bently Nevada Corporation
and
Tom Pfoh
Corporate Product Service Manager
Bently Nevada Corporation

Access to machinery condition information is essential to a company's success in today's competitive marketplace. Without machinery condition information, unscheduled or extended downtime can result.

For a machinery management program to achieve its goals, the information must be valid. This includes vibration, position, bearing and oil temperature information as well as process information, which is such an important part of the program. In addition, the basic machinery protection system must also be in excellent working condition.

There are four ways ~~to ensure~~ that these conditions are met:

1. Determine which data is required.
2. Install the instrumentation properly.
3. Ensure proper maintenance checks are performed.
4. Repair instrumentation promptly and correctly.

Determining what data is needed

The type of data that is required will depend on many factors including machine construction, application, importance to process, etc. For example, a machine with fluid-film bearings requires shaft vibration data obtained by proximity probes rather than bearing housing vibration data. And bearing temperature data, along with the vibration information, may give a more complete picture of the machine's condition. Our Sales Engineers and Machinery Diagnostic Services Engineers can assist in determining the appropriate measurements for your machinery. If the necessary transducers do not exist, however, they must be installed.

Proper installation

There are many factors to consider when installing transducers in order to ensure that they provide accurate data and have a long service life. These factors typically involve locating the transducer at the appropriate point for the measurement, mounting the transducer properly, and meeting the requirements of the physical limitations of the transducer. Our Design & Installation Services department has experience installing thousands of transducers in a wide range of applications.

Process data can be particularly helpful in evaluating machinery condition. The process data, however, must be combined with machinery data to be most effective. If the process data is available on a plant information or control network, a hardware and software communications link between the plant and machinery management systems must be established. This data can then be "integrated" to provide a better indication of machinery health. (See the article on page 20).

Performing proper maintenance

Once a system is properly installed and configured, it must be maintained to ensure proper operation. On a routine basis, the calibration and operation of the transducers, monitors, and machinery management system must be checked. To accomplish this, a technician must be knowledgeable about all aspects of system operation. In addition, it is important that the technician understands the data acquisition and data presentation aspects of the machinery diagnosis part of the management system. If a set of XY probes is mounted correctly, for example, but connected backwards, the data they provide can lead a machinery engineer to the incorrect conclusion about a machine's condition.

Prompt and correct repairs

If a failure occurs, it is important to promptly and correctly repair the system. This helps ensure that valuable data is not lost or unavailable when it is absolutely necessary to diagnose a machine malfunction. To do this, the technician must be familiar with the operation of the system down to the failed device level. Once the failed device is identified, the replacement part must be in stock to complete the repair.

Site Management Agreements

To ensure that systems are maintained and repaired properly, Bently Nevada offers Site Management Agreements. The goal of these agreements is to create a well-managed machinery management program that will reduce maintenance costs and catastrophic failures, increase machine availability, and extend machinery life.

The first step of the agreement is to make sure that the existing machinery protection system is operating correctly and can be relied upon. Bently Nevada's Product Service organization provides complete system checkouts and verification services to ensure the integrity of a system's operation.

During a system checkout, all required configuration updates and adjustments are made, so it will provide trouble-free operation. Once the system is providing accurate and reliable data, it will be remotely accessed to ensure the continued reliability of the system's hardware and software components. Computer-based machinery management systems provide faster response times to potential problems at a much lower cost than a site visit. As part of a Site Management Agreement, these services are scheduled on a regular basis and are included as part of a comprehensive management package.

Detailed reports are also provided at regular intervals to explain any corrective actions taken and any pending actions. In accordance with governmental regulations, any changes made to the machinery management system are documented along with alarm setpoints, calibration information, and serial numbers.

Having Bently Nevada perform the instrumentation maintenance as part of a site management program reduces your costs and increases the dependability of your machinery management system. Bently Nevada's Service organizations will perform any necessary maintenance and advise your personnel of the actions taken, in writing.

Our Machinery Diagnostic Services organization is part of a complete Site Management Agreement. Facilities with machinery management systems that are equipped with remote telecommunications can contact a qualified engineer, 24 hours a day, for advice on possible machinery problems. Upcoming Orbit articles will discuss our products' remote machinery diagnostic capabilities. ■

Continued from page 17 (Some realities of field balancing).

Dynamic Stiffness

Dynamic stiffness is the restraint of motion when a mechanical system is subjected to an oscillating or rotating force. This can be expressed in the general equation ***Motion = Force/Restraint.*** Dynamic stiffness is the sum of the stiffnesses associated with each of the physical characteristics of a system: spring, damping, and mass. Each of these stiffnesses results in a characteristic **lag** in the response of the system with respect to the applied force. This is most easily seen in the classical, one-dimensional spring-damper-mass model, (Figure 1).

In the rotating, two-dimensional rotor/bearing/seal system (Figure 2) each stiffness component, including that of the surrounding fluid, results in a distinct **direction** of response. The direction is associated with the phase lags seen in the one-dimensional case. When the system is unbalanced to provide an input force, the resulting motion is shown in Figure 2.

At low speeds, when the spring stiffness is dominant, the motion is in the direction of the unbalance force F (push on a spring and it moves in the direction of the force). It is determined by the ratio $\frac{F}{K}$, where K is the spring stiffness.

At resonance, when the spring and mass stiffnesses cancel, the motion lags the force by 90° (against the direction of rotation). The motion is in quadrature to the force and is determined by the ratio $\frac{F}{D(1-\lambda)\Omega}$, where $D(1\text{-}\lambda)\Omega$ † is the fluid wedge (tangential or cross-coupled) portion of the dynamic stiffness (push on a wedge and it moves at right angles to the force).

At high speeds, when mass stiffness is dominant, the motion lags the force by 180° (a track and field athlete swinging a hammer uses his mass to balance the system by moving away from the weight). The motion is determined by $\frac{F}{M\Omega^2}$, where $M\Omega^2$ is the mass portion of the dynamic stiffness.

The general expression for synchronous motion, then, is

$$Motion = \frac{Force}{Restraint} = \frac{Force}{\left(K + D(1\text{-}\lambda)\Omega - M\Omega^2\right)} .$$

† The general Quadrature Stiffness term is $K_Q = jD(\omega - \lambda\Omega)$, where ω is rotor precession rate and λ is the Fluid Circumferential Average Velocity Ratio due to the fluid dragged into rotation by the shaft, which reduces the effect of damping. In this case, $\omega = \Omega$, so that

$$K_{Q\,Synch} = D(1-\lambda)\Omega .$$

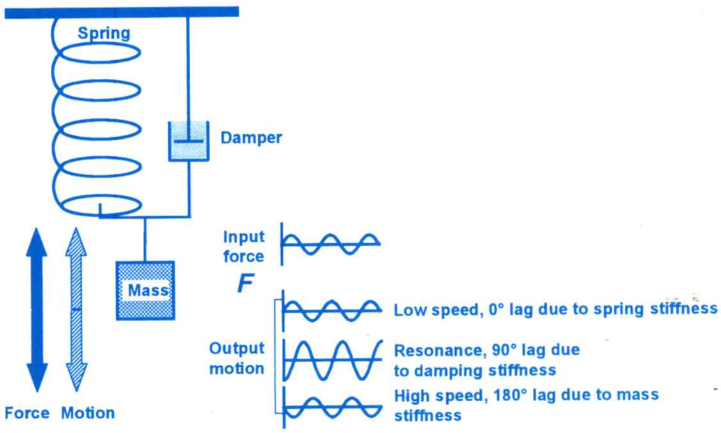

Spring
Damper
Mass
Force
Motion
Input force F
Output motion
Low speed, 0° lag due to spring stiffness
Resonance, 90° lag due to damping stiffness
High speed, 180° lag due to mass stiffness

Figure 1
Response of a one-dimensional spring-damper-mass system to an oscillating force.

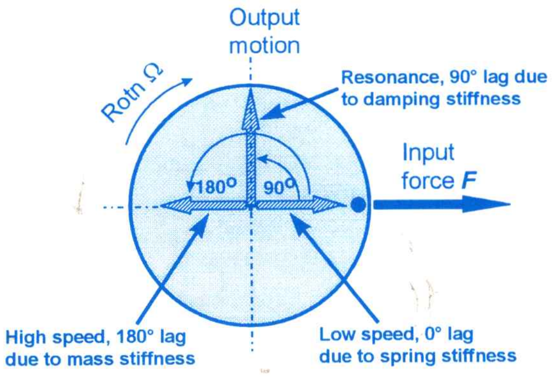

Output motion
Rotn Ω
Resonance, 90° lag due to damping stiffness
Input force F
180°
90°
High speed, 180° lag due to mass stiffness
Low speed, 0° lag due to spring stiffness

Figure 2
Response of a two-dimensional, rotating spring-damper-mass system to a force rotating Y to X (cw) at the same rate.

Detailed reports are also provided at regular intervals to explain any corrective actions taken and any pending actions. In accordance with governmental regulations, any changes made to the machinery management system are documented along with alarm setpoints, calibration information, and serial numbers.

Having Bently Nevada perform the instrumentation maintenance as part of a site management program reduces your costs and increases the dependability of your machinery management system. Bently Nevada's Service organizations will perform any necessary maintenance and advise your personnel of the actions taken, in writing.

Our Machinery Diagnostic Services organization is part of a complete Site Management Agreement. Facilities with machinery management systems that are equipped with remote telecommunications can contact a qualified engineer, 24 hours a day, for advice on possible machinery problems. Upcoming Orbit articles will discuss our products' remote machinery diagnostic capabilities.

Continued from page 17 (Some realities of field balancing).

Dynamic Stiffness

Dynamic stiffness is the restraint of motion when a mechanical system is subjected to an oscillating or rotating force. This can be expressed in the general equation ***Motion = Force/Restraint.*** Dynamic stiffness is the sum of the stiffnesses associated with each of the physical characteristics of a system: spring, damping, and mass. Each of these stiffnesses results in a characteristic **lag** in the response of the system with respect to the applied force. This is most easily seen in the classical, one-dimensional spring-damper-mass model, (Figure 1).

In the rotating, two-dimensional rotor/bearing/seal system (Figure 2) each stiffness component, including that of the surrounding fluid, results in a distinct **direction** of response. The direction is associated with the phase lags seen in the one-dimensional case. When the system is unbalanced to provide an input force, the resulting motion is shown in Figure 2.

At low speeds, when the spring stiffness is dominant, the motion is in the direction of the unbalance force F (push on a spring and it moves in the direction of the force). It is determined by the ratio $\frac{F}{K}$, where K is the spring stiffness.

At resonance, when the spring and mass stiffnesses cancel, the motion lags the force by 90° (against the direction of rotation). The motion is in quadrature to the force and is determined by the ratio $\frac{F}{D(1-\lambda)\Omega}$, where $D(1\text{-}\lambda)\Omega$ † is the fluid wedge (tangential or cross-coupled) portion of the dynamic stiffness (push on a wedge and it moves at right angles to the force).

At high speeds, when mass stiffness is dominant, the motion lags the force by 180° (a track and field athlete swinging a hammer uses his mass to balance the system by moving away from the weight). The motion is determined by $\frac{F}{M\Omega^2}$, where $M\Omega^2$ is the mass portion of the dynamic stiffness.

The general expression for synchronous motion, then, is

$$Motion = \frac{Force}{Restraint} = \frac{Force}{\left(K + D(1\text{-}\lambda)\Omega - M\Omega^2\right)}.$$

† The general Quadrature Stiffness term is $K_Q = jD(\omega - \lambda\Omega)$, where ω is rotor precession rate and λ is the Fluid Circumferential Average Velocity Ratio due to the fluid dragged into rotation by the shaft, which reduces the effect of damping. In this case, $\omega = \Omega$, so that

$$K_{Q\ Synch} = D(1-\lambda)\Omega\ .$$

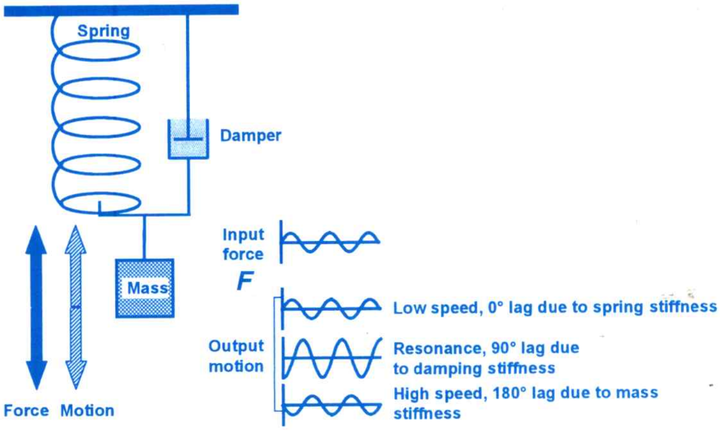

Spring
Damper
Mass
Force
Motion
Input force
F
Output motion
Low speed, 0° lag due to spring stiffness
Resonance, 90° lag due to damping stiffness
High speed, 180° lag due to mass stiffness

Figure 1

Response of a one-dimensional spring-damper-mass system to an oscillating force.

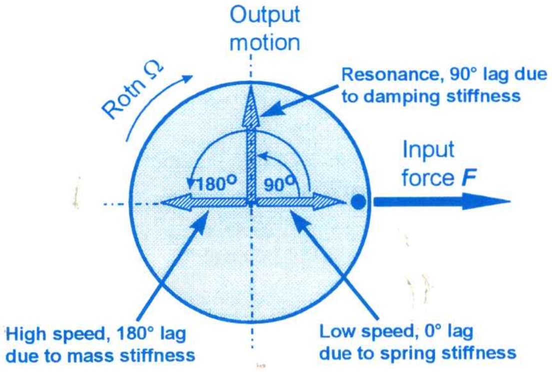

Output motion
Rotn Ω
Resonance, 90° lag due to damping stiffness
Input force F
180°
90°
High speed, 180° lag due to mass stiffness
Low speed, 0° lag due to spring stiffness

Figure 2

Response of a two-dimensional, rotating spring-damper-mass system to a force rotating Y to X (cw) at the same rate.

System Connections

Case history

System integration prepares Southern California Edison for future power demands

by Dan Wadelton, P.E.
Corporate Product Service
Support Engineer
Bently Nevada Corporation

Two computer systems vital to the operation and health of Southern California Edison's (SCE) El Segundo Power Plant couldn't communicate with each other. The result was potentially costly. Operators and Machinery Specialists needed the information from both systems to ensure that the plant ran smoothly, and they could not get to it simply. The imminent deregulation of the electric utility industry and the need for maximum utilization of existing generating equipment made the shared database even more important.

In a proactive move, SCE commissioned Bently Nevada's System Integration Services group to interface those two systems, Bently Nevada's Data Manager® 2000 for Windows NT™ System (DM2000) and the Bailey® Infi 90 Distributed Control System (DCS), at their plant in El Segundo, California.

The project objectives were:

1. To enable the Bently Nevada DM2000 system to access the process data resident on the Bailey system. This allows the Machinery Specialist to correlate the process data with the vibration data to facilitate problem diagnosis.
2. To enable the Bailey Operator Interface Stations (OIS) to access the Bently Nevada Data Manager System as remote terminals. This allows the Operators to view all the information critical to their decisions.
3. To permit the Machinery Specialist to access the DM2000 and its expanded database from any location, internal or external to the plant.

Southern California Edison's El Segundo Power Plant

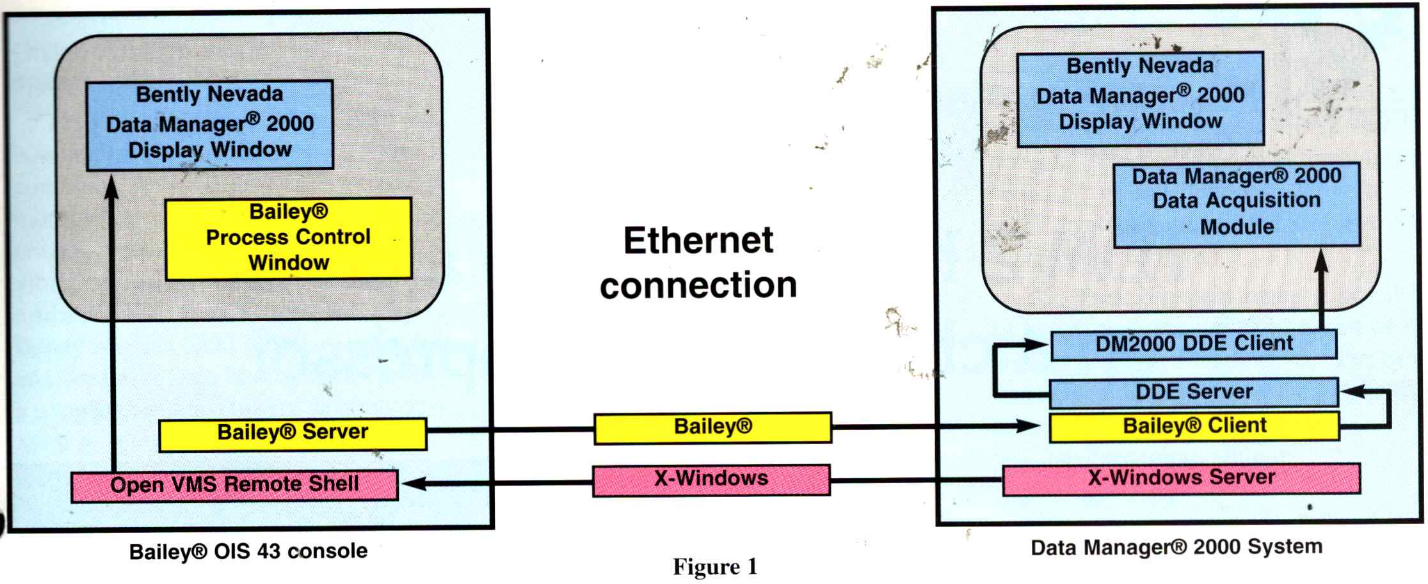
Photos courtesy of Southern California Edison

Getting Data Manager 2000 to access INFI 90 OPEN data

The existing Bailey DCS installation is made up of seven OIS-42 Series control consoles on an Infi 90 Open super loop. One console had recently been upgraded, which enabled it to act as a data server.

The Bently Nevada DM2000 Software, designed to run on a Microsoft Windows NT platform, can bring data into its database via a Dynamic Data Exchange (DDE) server. By adding a separate Bailey client package to the DM2000 computer, we provided a way for the DDE server to access the Bailey INFI 90 OPEN data. This could only happen, however, if the systems could physically talk to each other. The network connection was the next link in the operation.

The network link was accomplished by first physically connecting the systems to the same Ethernet, Wide Area Network (WAN). Then, after configuring the protocols on each system, the Bailey client software on the DM2000 was able to access the real-time process data on the INFI 90 OPEN database. The next step was to activate the DDE Server and configure the DM2000 Software with the appropriate tag names of the data it needed. Process data could now be cor-



Bently Nevada Data Manager® 2000 Display Window
Bailey® Process Control Window
Bailey® Server
Open VMS Remote Shell
Bailey® OIS 43 console
Ethernet connection
Bailey®
X-Windows
Bently Nevada Data Manager® 2000 Display Window
Data Manager® 2000 Data Acquisition Module
DM2000 DDE Client
DDE Server
Bailey® Client
X-Windows Server
Data Manager® 2000 System

Figure 1
System block diagram.

related with vibration data in the Data Manager® 2000 Display software module. We had now met our first project objective.

At the same time, we met our third project objective and significantly increased the effectiveness and decreased the response time of the Machinery Specialist when he needed to diagnose potential machinery problems. He could now access the DM2000 directly, through the WAN, or by telephone through the Remote Access Services capabilities of Windows NT. Since the Display module can run separately from the Data Acquisition module, the Machinery Specialist could access vibration and process data anytime, from anywhere, with his own computer, running Windows 95.

Data Manager 2000 display on Bailey consoles

Providing Operators with the capability to run the DM2000 display module on their consoles was a more complex problem. The consoles ran a different operating system, Open VMS, that did not support our software or Microsoft's. Open VMS does support X-Windows, however. X-Windows, or simply "X," is a widely-adopted, graphical windowing system which is hardware and software independent. X could display the screens from the DM2000 Display module. Unfortunately, X is not standard with Microsoft.

However, a third party X-Windows Application Server could be installed to run under Windows NT on the DM2000 computer. When the operator clicked on a standard icon on his Bailey console, it would run a remote shell command and automatically log on to the DM2000 System. Now, the Operator could launch the Data Manager 2000 Display module and, thanks to X, have the exact same graphical user interface that he would have if he were sitting at the DM2000 computer. This completed project objective two, giving the Operator all the information available that is critical to his decision-making process.

System integration, using Data Manager® 2000, provides operators at the El Segundo Power Plant with information critical to their decision-making process.

Integration Services

Bently Nevada's System Integration Services was instrumental in helping Southern California Edison along the path to achieving maximum utilization of their resources and a more secure business future. If you would like to consider a similar installation using Data Manager 2000 and your process control system, contact your sales representative and ask them for information regarding our System Integration Services. Our staff welcomes the opportunity to discuss your particular needs and to share new ideas. ■

Machinery Messages

Case history

TDM 2 helps diagnose a crack on a charge gas compressor

by *Fabio H. G. Martins*
Maintenance Engineer
Petroquimica União S.A.
and
Rolando G. Dias
Manager/MDS Engineer
Bently do Brasil Ltda.

Aerial view of Petroquimica União S.A., Santo André, São Paulo, Brazil.

Petroquimica União S.A. (PQU) is a petrochemical plant located in Santo André, São Paulo, Brazil, that produces basic petrochemicals such as ethylene, propylene, butadiene, benzene, etc. These chemicals are raw materials for second and third generation industries, and are further transformed into many plastic products.

The machine train

GB-201, a 32,000 hp centrifugal compressor operating at the beginning of the process line, compresses a mixture of gases, called "charge gas," from the naphtha cracking furnaces. There are five compressor stages contained in three casings: low, medium, and high pressure.

The couplings are gear type, so the power is transmitted between rotors by gear teeth. Small axial displacements, due to load variation and thermal expan-

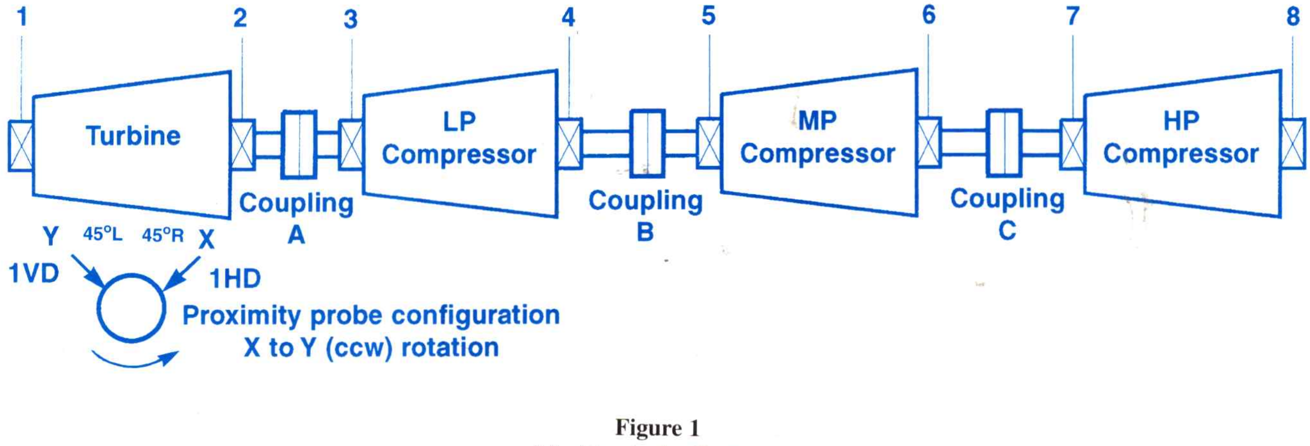

1
2
3
4
5
6
7
8
Turbine
Coupling A
LP Compressor
Coupling B
MP Compressor
Coupling C
HP Compressor
Y
45°L
45°R
X
1VD
1HD
Proximity probe configuration
X to Y (ccw) rotation

Figure 1
Machine train diagram.

sion, as well as residual misalignment, are absorbed by the couplings.

Online vibration monitoring system

The operational reliability of GB-201 is essential for the process, so PQU is continuously investing in systems to monitor the condition of this and other critical machines. In 1994, the GB-201 vibration monitoring system was modernized. The installation included a Bently Nevada 3300 Monitoring System and Transient Data Manager®2 (TDM2), a computerized machinery data management system.

This system allows for simultaneous and continuous data collection from all sensors. Data is periodically sent to a remote computer, where all vibration characteristics may be stored and analyzed during steady state or transient (startup/shutdown) operation. The system generates plots that include:

- Orbits and waveforms (direct and filtered)
- Spectrum & spectrum cascade
- Bode and polar
- Trends of overall and filtered signals, including phase lag angle
- Shaft centerline position

Occurrences

After the last overhaul in 1994, compressor vibration levels had been quite steady and acceptable. However, in late December 1995 and early January 1996, the plant experienced two shutdowns due to operational problems. The compressor was first stopped and then had its speed reduced. After these events, a sudden change was observed in overall vibration levels (Figure 2) in bearing #4.

The problem was diagnosed as a coupling B lockup because the vibration changes were sudden. There was evidence of poor coupling lubrication and excess water, probably due to leakage in the turbine seals. It was recommended that a close watch be maintained. If vibration levels suddenly increased, the machine would be shut down and started up again. The procedure was to attempt a "release" of the coupling, through the axial displacement of shafts during the shutdown and startup.

The TDM2 system made it possible to better evaluate the characteristics of another event in March 1996. The system helped identify the real cause of the sudden increase in vibration and avoid a possible catastrophic failure.

Crack diagnosis

The observation of vibration trends after January 1996, showed a continuous increase of 1X vibration in bearing #2, but with a steady phase lag angle. The overall vibration level stabilized at

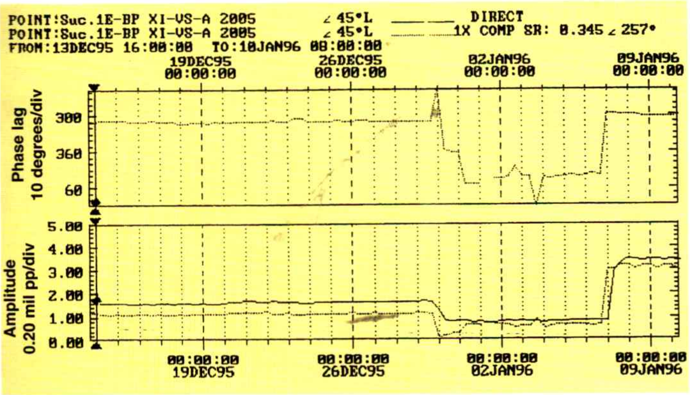


Figure 2
Bearing #4 vibration trend plot (1X) between 13 December 1995 and 10 January 1996.

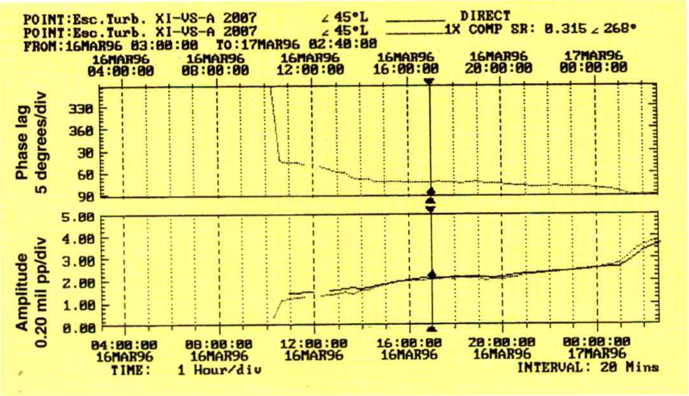


Figure 3
Bearing #2 vibration trend plot (1X). 16 to 17 March 1996.

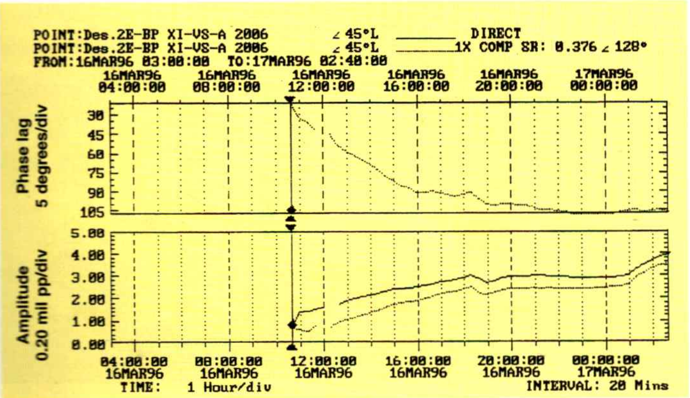


Figure 4
Bearing #3 vibration trend plot (1X). 16 to 17 March 1996.

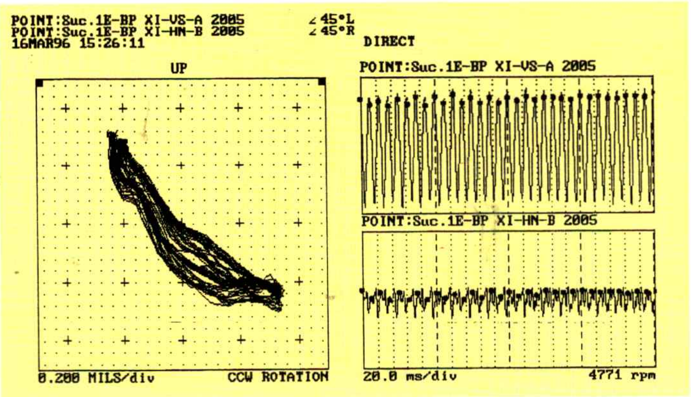


Figure 5
Bearing #4 orbit/timebase plot. 16 March 1996.

approximately 25µm (1.0 mil), and no further action was taken.

On March 16th, however, after a boiler shutdown caused an unexpected steam supply upset, fast vibration changes were observed at bearings #2, #3, and #4. At first, we assumed it was another coupling lockup and tried to release it by varying the compressor load and speed. The vibration levels increased, however, and exhibited the following characteristics:

- Predominant 1X vibration, continuously increasing.
- Continuous 1X vibration phase shift (Figures 3 and 4).
- 2X vibration at bearings #2 and #3, suggesting shaft stiffness asymmetry associated with a constant radial load.
- Banana-shaped orbit at bearing #4 (Figure 5), indicating excessive load, possibly due to gross misalignment or coupling lockup, due to poor lubrication.

The diagnosis (based on this data and a history of shaft cracks in this machine) was that there was a crack in the rotor of the compressor low pressure stage. (The low pressure compressor shaft had been replaced twice before, in 1989 and 1992, due to cracks. Both times, these cracks occurred in the keyway of the coupling hub. A hydraulically-assembled, diaphragm coupling had been installed in July 1996 to alleviate this problem.) The plant was then shut down for the necessary inspection and maintenance.

Inspection

The inspection did find a crack, but in a different component, coupling A. There was a crack through nearly 60% of the coupling section (Figure 6 and 7). A fracture analysis carried out on coupling A showed that the crack was due to fatigue, starting at three different spots.

Coupling B lockup was also confirmed, due to a large amount of oil sludge inside. The same condition was found in couplings A and C.

Root cause analysis

Further evaluation lead to the following conclusions:

- Lubrication oil was contaminated with water from a turbine seal leakage, producing oil sludge.
- There was sludge build-up around the coupling teeth, due to centrifugal forces.
- The coupling had locked up and began to act as a rigid body, unable to compensate for the residual misalignment.
- The coupling failed, due to fatigue caused by cyclic forces.

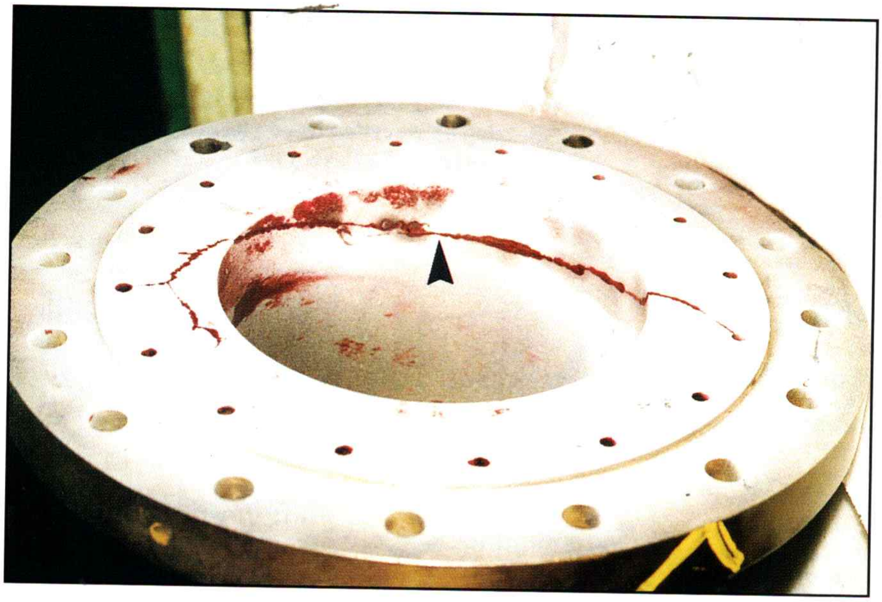
Conclusion

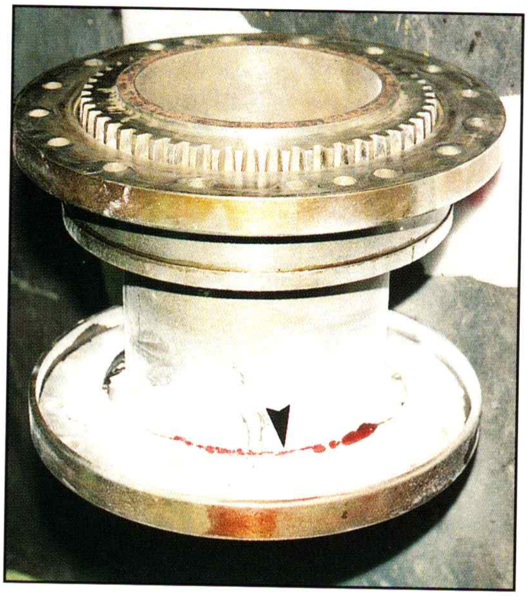
The information lead to two straightforward conclusions:

- A major failure was avoided because the right decision was taken, based on high-level, readily available information.
- The Bently Nevada TDM2 System was essential in providing engineers both appropriate and accurate information so they could identify the problem. ■

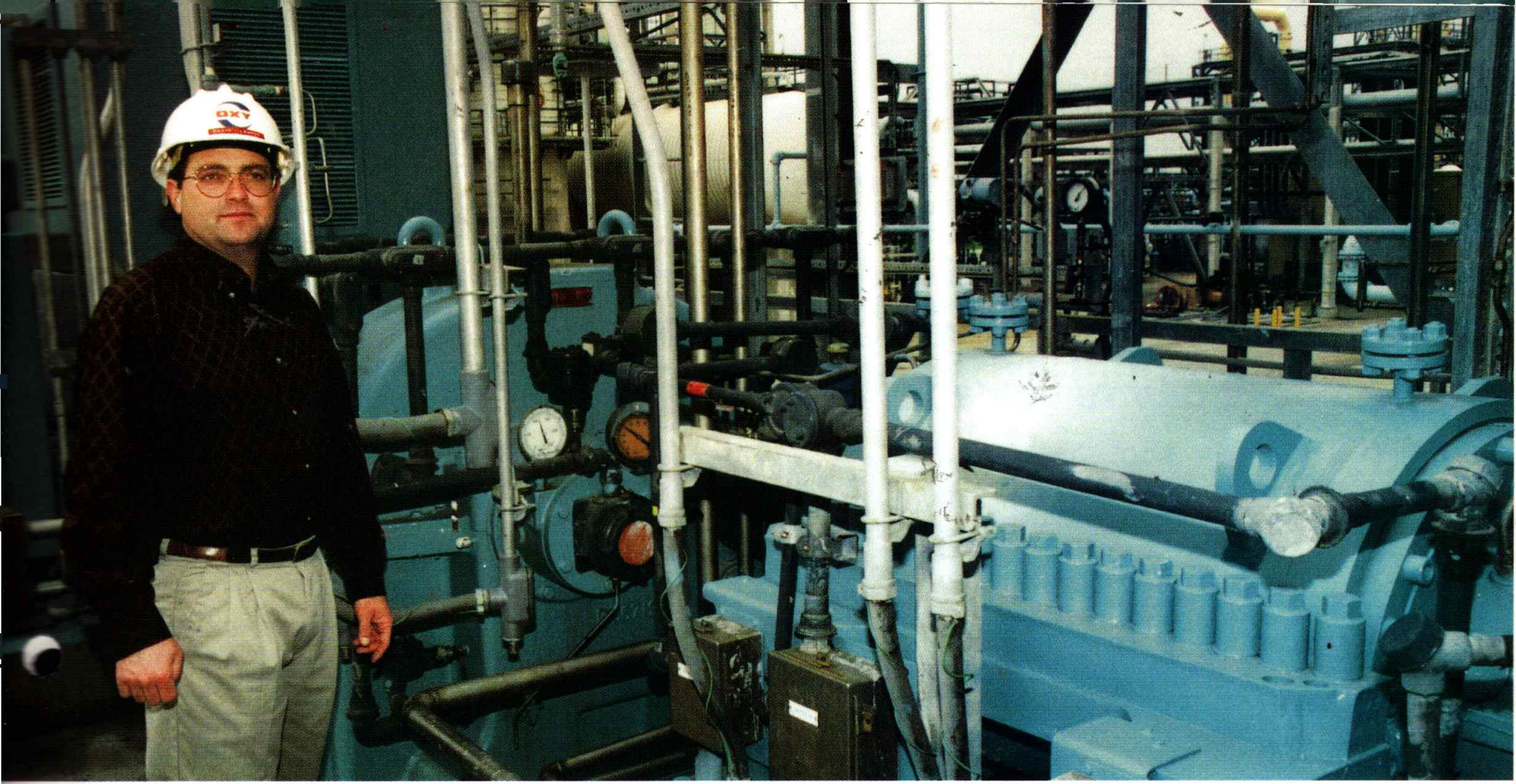
Editor's Note: Two of the main shaft crack symptoms are 1X vibration change and phase shift. Note that after 20:00, 16 Mar 96, (Figures 3 and 4) the phase remained quite constant. One cause of the continuous phase shift during a shaft crack is the change in the radial direction of the lower shaft stiffness, as the crack propagates. It was already getting close to 50% of the coupling section, so the direction of lower shaft stiffness remained about the same. It is reasonable to conclude that, when the phase stopped shifting, the crack process did not stop. In this case, the crack reached 60% of the coupling section.

Photos courtesy of Petroquimica União S.A.





Figures 6 and 7
Internal and external view of cracked coupling. Arrow indicates location of the crack.



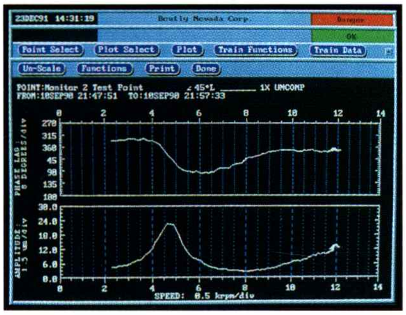
“The Transient Data Manager® 2 System plays a key role in the way we manage our machines.”

David Lawhon
Rotating Equipment Engineer
Occidental Chemical Corporation
Ingleside, Texas USA

Transient Data Manager® 2 is installed at Occidental Chemical’s Ingleside plant. The machinery data management system acquires their critical machinery data and plots the information in several powerful formats. Plant personnel have immediate confirmation that the machinery is running safely and within specification.

By protecting and managing their assets, Occidental improves plant productivity, availability and can better utilize plant personnel for other tasks. The machinery data management system assists Occidental in avoiding a shutdown while improving their bottom line.

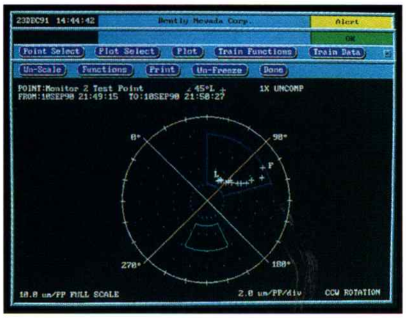
Call your Bently Nevada sales office for a comprehensive range of machinery management systems designed for critical, essential, and general-purpose machinery.



23DEC91 14:31:19
Bently Nevada Corp.
Danger
OK
Point Select
Plot Select
Plot
Train Functions
Train Data
Un-Scale
Functions
Print
Done
PHASE LAG: 0 DEGREES/div
AMPLITUDE: 1.5 um/div
SPEED: 0.5 krpm/div

```
POINT:Monitor 2 Test Point         ∠45°L _______ 1X UNCOMP
FROM:10SEP90 21:47:51  TO:10SEP90 21:57:33
```

Display critical information during startup and coastdown



23DEC91 14:44:42
Bently Nevada Corp.
Alert
OK
Point Select
Plot Select
Plot
Train Functions
Train Data
Un-Scale
Functions
Print
Un-Freeze
Done
0°
90°
270°
180°
F
L
10.0 um/PP FULL SCALE
2.0 um/PP/div
CCW ROTATION

```
POINT:Monitor 2 Test Point         ∠45°L +        1X UNCOMP
FROM:10SEP90 21:49:15  TO:10SEP90 21:50:27
```

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BENTLY NEVADA®

New 3500/45 Position Monitor can be a thrust, differential expansion or case expansion monitor

by Greg Paulson
Marketing Specialist
Bently Nevada Corporation

The 3500/45 Position Monitor now provides inputs for three critical position parameters: thrust, differential expansion, and case expansion. These three parameters provide an accurate picture of the axial position condition of your steam turbine.

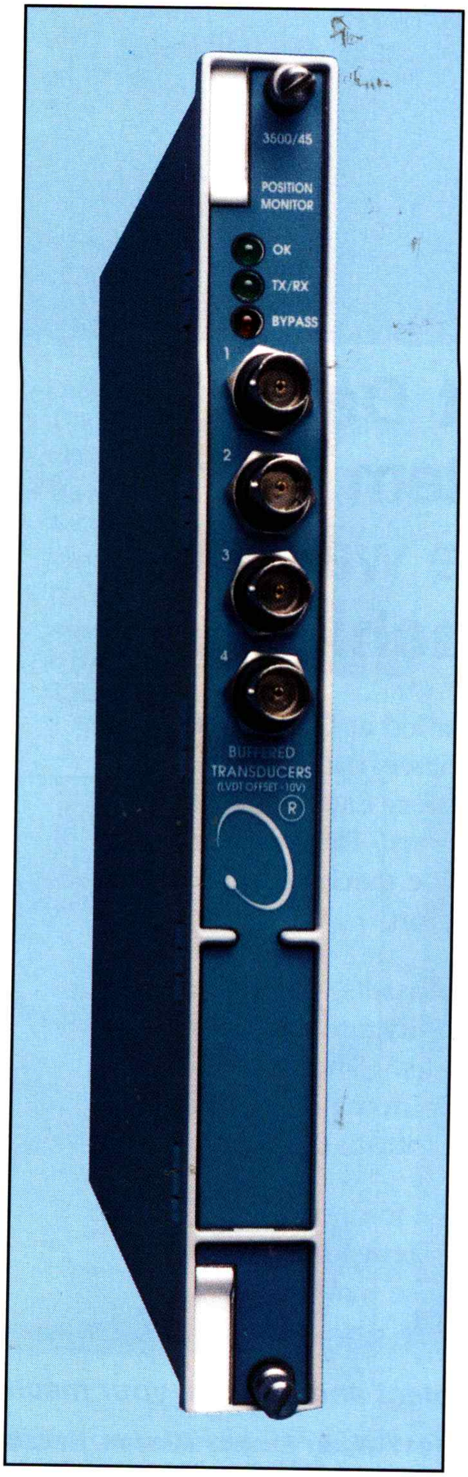
The rotor and case of large steam turbines grow at different rates during startup, shutdown, and load changes. Monitoring this growth is necessary, as rotating parts can contact stationary parts of the steam turbine, causing substantial machine damage. Typically, case expansion and differential expansion are used to monitor this growth.

Differential expansion

Differential expansion measurements compare the growth of the rotor relative to the case. As the machine heats, the rotor may expand faster than the case, or the case may expand faster than the rotor, depending upon the rotor-to-case mass ratio. For example, if the rotor has less mass than the case, it will expand faster than the case. By measuring rotor growth relative to the case, limits can be set to indicate when the turbine must be stabilized (heat soaked). This allows the differential growth between the case and rotor to decrease before increasing the temperature.

Case expansion

Case expansion is a measurement of machine case growth relative to its foundation. During the startup of a steam turbine, case growth is expected. Abnormal case growth occurs when a case foot becomes stuck, which can cause the case to expand unevenly. A stuck foot tends to misalign the rotor and increase case stress which can lead to a rub or excessive vibration.



A case expansion monitor protects a turbine from damage from abnormal case growth by warning an operator when one side of the case grows more than the other. In an ideal application of case expansion monitoring, two linear variable differential transformers (LVDTs) are used, one on each side of the case. Alarms can be set, based on predefined differences between the two LVDT measurements. This will warn the operator that the case foot is stuck and must be corrected. The operator will typically inspect the foundation, make sure it is properly greased, and ensure no foreign objects are in the expansion slots prior to using a jack or hammer to free the case foot, so the case can move freely.

3500/45 Position Monitor

The 3500/45 Position Monitor is an enhancement to the original 3500/45 Differential Expansion/Thrust Monitor. It incorporates the case expansion measurement. This four channel monitor is easy to set up, using our 3500 Configuration Software. Since one monitor can serve as a spare for different monitor types, spare parts costs can be reduced. Any channel in a 3500/45 can be configured to monitor thrust position or differential expansion. Several different types of differential expansion measurements are supported including:

- Standard Differential Expansion
- Complementary Input Differential Expansion (CIDE)
- Standard Single Ramp Differential Expansion
- Nonstandard Single Ramp Differential Expansion
- Dual Ramp Differential Expansion

The differential expansion and thrust channels of the Position Monitor will accept signal inputs from the following Bently Nevada transducers:

- 5, 8, 11, and 14 mm 7200 Systems
- 8 mm 3300 System
- 25, 35, and 50 mm Extended Range Systems
- 50 mm Differential Expansion Transducer
- 3300 RAM System
- 3000 Series System

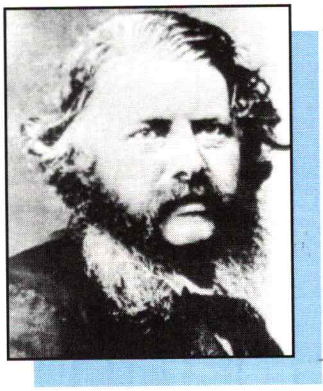
The Position Monitor can be configured for case expansion measurement on Channels 3 and 4 and will accept signal inputs from the existing Bently Nevada 24765 Case Expansion Transducer. Bently Nevada will soon manufacture a new High Temperature Case Expansion Transducer, which will have CE approval, an improved operating temperature of 85°C (185°F), and will be compatible with the Position Monitor.

The Position Monitor provides system flexibility and optimum channel utilization. Differential expansion measurements can be made on Channels 1 and 2 and case expansion measurements on Channels 3 and 4. If the turbine requires more than two channels of differential expansion measurements, Channels 3 and 4 can also be used for differential expansion, and the case expansion measurements can be moved to another Position Monitor. Any channels not used for differential expansion or case expansion can be configured for thrust measurements.

The 3500/45 Position Monitor is a component of Bently Nevada's new 3500 Machinery Data Management System. Contact your nearest sales representative for more information. ■

Famous names in rotor dynamics

William Rankine introduced much of the terminology and notation of thermodynamics still in use today



William J. Macquorn Rankine ('rang kin), 1820-1872, a "Renaissance" man dedicated to translating theoretical principles into practical terms in the Steam Age, may have also been responsible for setting back the science of rotor dynamics by nearly fifty years.

Rankine was born in Edinburgh, Scotland. He was the son of an army lieutenant and received his early schooling from his father. Although he was a successful student, trained in physics, he left the university before obtaining a degree. For four years, he worked as an apprentice on railroad and hydraulic projects, occasionally collaborating with his father. This experience seems to have gotten him interested in the study of heat and heat engines, and probably reinforced his drive to translate the remotely theoretical into the eminently practical.

He first started publishing papers on physical sciences topics in 1849. He ultimately produced a set of comprehensive engineering textbooks, including, "A Manual of the Steam Engine and Other Prime Movers," 1859, which brought thermodynamics to working engineers. In 1869, Rankine published the first work on the dynamics of rotating shafts, "Centrifugal Whirling of Shafts," Engineer, XXVI, April 9, 1869. This became a classical starting point of the new discipline known today as Rotor Dynamics, and ultimately, Rotating Machinery Dynamics.

Rankine was a professor at the University of Glasgow from 1855 until his death, a consultant, and a highly respected authority in a wide range of engineering and scientific topics. He was a bold thinker who freely invented terminology for the disciplines he was interested in. He introduced much of the terminology and notation of thermodynamics, most of which is still in use today. He coined the terms "potential energy" and "critical speed" among others.

It was his boldness and willingness to venture where others hadn't that had a negative effect on rotor dynamics for many years. Rankine's statement in 1865 that, "For every length of shaft there is a speed that cannot be exceeded, and for every speed there is a length of shaft that cannot be exceeded." was the first description associated with limits in machine operation. Though the rule is generally true, what he actually meant was that the first balance resonance was the speed limit for machine operation; this was incorrect. Unfortunately, the influence of this rule lasted over thirty years, even though de Laval showed that machines could run above their "critical" (balance resonance) speed in 1883. ■

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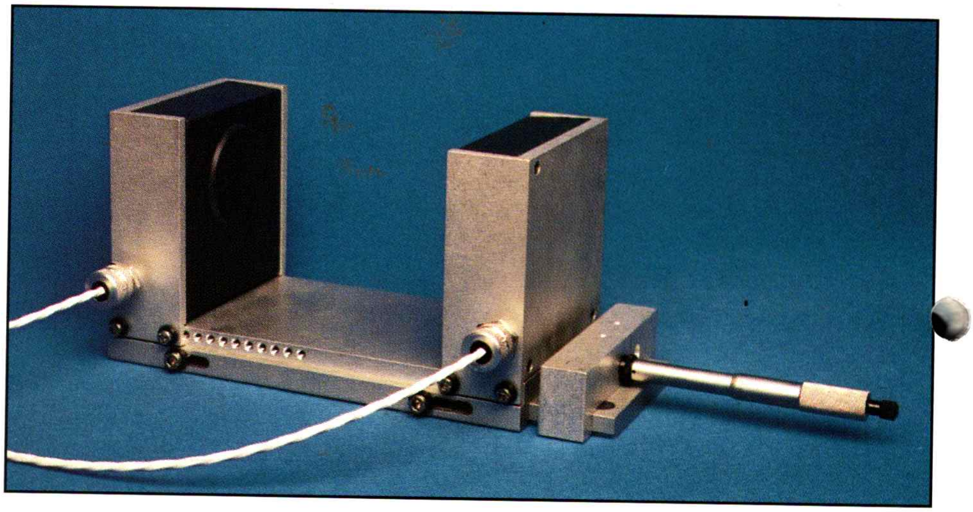
Accurate and reliable differential expansion measurements

by Mike Hanifan
Marketing Specialist, Large Machines
and Reciprocating Compressors
Bently Nevada Corporation



50 mm Differential Expansion Transducer mounting bracket with probes in a complementary input configuration.

The 50 mm Differential Expansion (DE) Transducer has the *longest linear range* of any non-contacting transducer system produced by Bently Nevada to date. Its advanced electronics, gives it a linear range from 1.3 mm (0.05 inches) to 29.2 mm (1.15 inches). If mounted using the complementary input configuration described below, its maximum range doubles to 58.4 mm (2.3 inches). The 50 mm DE Transducer also provides you with superior accuracy, reliability, ease-of-installation, and verification. It is an excellent solution for measuring differential expansion on large steam turbines.

The 50 mm Differential Expansion Transducer is our most accurate and reliable transducer for proximity measurements where the maximum range exceeds 12.5 mm (0.49 in).

Differential expansion

Most steam turbines, especially large turbines, undergo differential expansion under transient conditions. Differential expansion is the change in axial position of the rotor relative to the casing, due to thermal expansion, which occurs primarily during machine startup and shutdown.

By closely monitoring the position of the rotor relative to the machine casing (at some distance from the thrust bearing), you can ensure that the difference in expansion rates between the rotor and the casing remains within specified limits. By incorporating differential expansion into an Emergency Shutdown System (ESS), you can significantly reduce or eliminate damage from axial rubs. Most Turbine Supervisory Instrumentation (TSI) systems include at least one differential expansion measurement.

The two most common ways to measure differential expansion both use proximity transducers. In one system, the transducers observe a straight collar on the turbine rotor. In the other system, the transducers observe a ramp on the rotor.

The straight collar system uses one proximity transducer, mounted on the casing, to observe the axial movement of the collar. If the range of the differential expansion movement is greater than the range of the proximity probe, then two probes are installed in a Complementary Input for Differential Expansion (CIDE) configuration. In the CIDE configuration, two probes monitor the movement of one or two collars by facing the collar(s) in opposite directions (Figure 1). This doubles the measurement range since, as the collar extends beyond the range of one proximity probe, it moves into range of the other.

The other common differential expansion measurement uses two proximity probes, mounted to the case, to observe ramps machined into the rotor (Figure 2). The monitoring system converts the proximity probe range (measured perpendicular to the ramp surface) into an equivalent axial movement. Unlike the straight collar complementary input configuration, two transducers do not increase the measurement range. Axial movement of the rotor produces a signal which is opposite in polarity for the two proximity probe transducers. Radial movement of the rotor produces a signal of the same polarity for the two proximity probe transducers. By subtracting the signals from the two transducers to compensate for this radial movement, you can determine the magnitude of the differential expansion in the axial direction.

Improved installation and verification

During installation and maintenance overhauls, the correct operation of the transducer system should be verified and the appropriate gap (distance) set between the transducer and the target. The proximity transducer senses the distance between the probe tip and an observed surface. Eddy currents are generated in the target surface and a change in the strength of the return signal is detected by the transducers. Verification simulates the motion of the shaft collar.

It is accomplished by physically moving the transducers and recording outputs across the entire transducer range.

The probes must also be moved when setting the gap. When a probe is gapped, you adjust the distance between the probe and the observed surface to the distance you want the probe to be from the target under startup conditions.

Enhanced mounting

In the past, gapping and verifying differential expansion transducers has been difficult, especially for those used in a complementary input configuration. Most DE mounting brackets are fabricated so the bracket cannot be moved relative to the case. If the bracket which holds the DE transducers cannot move, the transducers must be either mechanically gapped in place or removed from the bracket and moved by hand. Both methods introduce errors in the verification and are time-consuming.

Improved mounting bracket

To solve problems associated with gapping and verifying DE transducers, Bently Nevada developed an *enhanced mounting bracket* for the 50 mm DE transducer. This optional bracket allows the transducers to move, so gapping and transducer verification is quick, easy, and accurate. The transducers mount on a bracket that slides on a stationary base. During operation, the sliding mounting bracket is secured to the stationary base to prevent movement. We designed brackets to use in three separate applications: a single transducer mounting, a complementary input configuration for viewing a straight collar, and a complementary input configuration viewing straight couplings.

Simplified Design

Most Bently Nevada eddy current transducer systems have separate electronics for the Proximitor® sensors and proximity probes. The new 50 mm DE Transducer system combines an eddy-current Proximitor® sensor and probe in one sensing unit. This more basic, integral design provides the following advantages:

- *Simpler installation.* You now need only one mount, since the Proximitor® sensor and the probe are now combined.

- *Fewer spare parts are required.* This design eliminates the need for a cable from the probe to the Proximitor® sensor. Therefore, fewer spare parts are necessary in your inventory.

- *Easier to troubleshoot.* Since there are fewer components and connections, it takes less time and effort to determine the root cause of a problem.

High accuracy

During the design of the 50 mm DE Transducer, we concentrated on increasing the accuracy of our differential expansion measurement. We reduced errors by 75% over our previous 50 mm Extended Range Transducer by introducing two major improvements:

- The standard new 50 mm Transducer is a matched system which is calibrated at the factory as a system. We eliminated the cable between the Proximitor® sensor and probe, and any errors normally introduced by this cable.

- A new *on-board temperature sensor* compensates the signal over the wide temperature range of 0° C to 120° C (32° F to 248° F). The compensated output provides more accurate differential expansion measurements throughout the operating temperature range.

Value for your investment

- The 50 mm DE Transducer maintains its full linear range for collar heights as low as 101.6 mm (4 in). It has a 25.4 mm (1 in) linear range on collars as low as 88.9 mm (3.5 in) without modifications. In most cases, we can modify the 50 mm DE Transducer or the monitoring system for use on collars as small as 63.5 mm (2.5 in) with a decrease in range.

- The transducer comes with a FluidLoc™ cable, which *inhibits oil from leaking* through the cable housing and out of the machine.

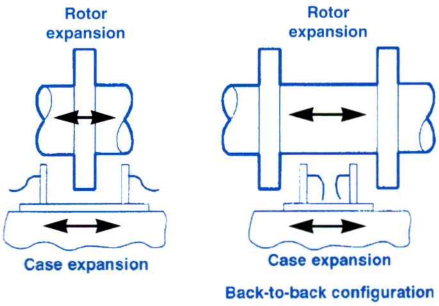
- A shield built into each transducer *reduces sideview* of the rotor surface.

- The 50 mm DE Transducer is a *cost-effective solution* for your differential expansion needs. Our improved 50mm DE Transducer sells for the same price as our previous 50 mm Extended Range Proximity System. With less expensive installation and verification costs, it provides a greater return on your investment.

- The 50 mm DE Transducer is *compatible* with our existing 7200, 3300, and 3500 Differential Expansion Monitors.

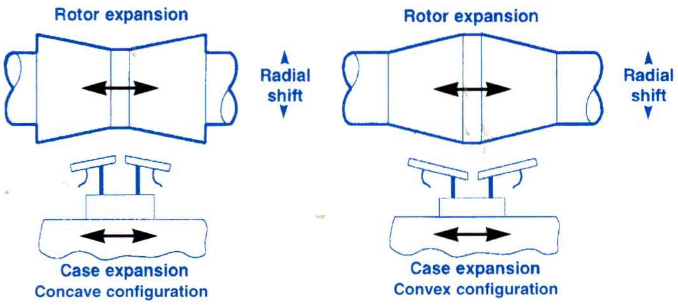
- After rigorous testing, the 50 mm DE Transducer was found to be compliant with Directive 89/336/EEC (EMC Directive) and received the *European CE Mark.*

For more information, contact your nearest Bently Nevada sales and service representative. ■



Back-to-back configuration

Figure 1
50 mm DE transducers in a Complementary Input for Differential Expansion (CIDE) application.



Concave configuration

Convex configuration

Figure 2
50 mm DE transducers in a ramp configuration.

ADRE® for Windows–now more versatile and powerful

Essential machinery information for fast and accurate machinery diagnostics

BENTLY NEVADA
MADE IN U.S.A.
Power
ADRE®
208-P Data Acquisition Interface Unit
Enhanced Parallel Port Communications

by Wicher Pot
Marketing Specialist
Bently Nevada Corporation

The world's most powerful portable diagnostic instrument, ADRE® for Windows, has become more versatile and more powerful. It now runs on many more types of notebook computers, automatically performs calculations that make diagnostics easier and more accurate, and lets you apply meaningful labels to certain plots.

Essential when problems occur

An advanced machinery data management system, such as Bently Nevada's Data Manager® 2000 for Windows NT™, automatically collects and processes machine vibration and process data. Changes in that data indicate conditions that could lead to premature failure and machine faults at an early stage of development. Once a fault has progressed, the data identifies the nature, severity, and location of the fault.

Machines that are not connected to a continuous online machinery data management system can be investigated with the same degree of sophistication, with Bently Nevada's ADRE for Windows System.

ADRE for Windows is a compact, lightweight diagnostic instrument that a machinery diagnostics engineer carries to the site of a suspected machinery problem. He connects it to the machine's transducers or monitoring system, and it immediately begins collecting and analyzing data from several transducers, simultaneously. The engineer can view data as it is collected, store it for later analysis, or both. He can view data from several transducers, each in its own "window," view data from one transducer in several different plot formats, each in its own window, or both. If the problem occurs infrequently, he can leave the system in place and set it to automatically collect data.

The ADRE for Windows System consists of software, a computer that runs the software, and hardware that collects machinery data. The data collection hardware, the Data Acquisition Interface Unit (DAIU) is now more versatile and powerful.

Works with wider variety of notebook and other computers

Now, ADRE for Windows does not require a special adapter to interface with your computer. It uses the computer's Enhanced Parallel Port (EPP). The new 208-P DAIU (P indicates the EPP interface) can be connected to a wider variety of notebook & desktop computers.

More power, easier to use

ADRE for Windows Software has also been significantly improved. Version 3.1 works with the 208 DAIU and the 208-P DAIU, and it automatically detects which DAIU is connected. Version 3.1 also runs under Windows 95. The new ADRE for Windows System makes it easier to analyze machine performance, and gives you more powerful tools to use. Its most important improvements are:

- Easier capture of large amounts of data
- Flexible vector math calculations
- Quick and easy polar plot rescaling
- Better indication of "split resonances"
- More informative spectrum plots

Easier capture of large amounts of data

Now, ADRE for Windows saves travel time for machinery specialists who collect an extensive amount of machine data. When ADRE for Windows fills its database with machinery data, it can automatically store the database, reinitialize itself, and begin collecting more data – without operator intervention.

ADRE for Windows is essential for detecting problems that occur infrequently. Simply connect the system to the machine's transducers or monitoring system, specify events that will activate data collection, and walk away. ADRE for Windows automatically collects data when it detects any of several user-specified events, such as a change in machine speed, amplitudes (1X, 2X, nX or direct vibration, probe gap, or process variable), phase lag, or a closing switch contact.

Flexible vector math calculations

Shaft absolute vibration is an important quantity that, on certain machines, is necessary information when evaluating dynamic responses caused by unbalance. It is useful for machines that transmit a significant amount of vibration to bearing pedestals or machine casings, typically machines that have soft supports or a high rotor-to-stator mass ratio.

Shaft absolute vibration is calculated by the vector addition of two different signals that ADRE for Windows has already collected. One is a proximity probe signal, which indicates the motion of the shaft relative to a bearing housing. The other is a velocity or Velomitor® signal, which shows the motion of the bearing housing relative to a fixed reference. ADRE for Windows can now perform the calculations, and present the results in an easy-to-understand format.

Quick and easy plot rescaling

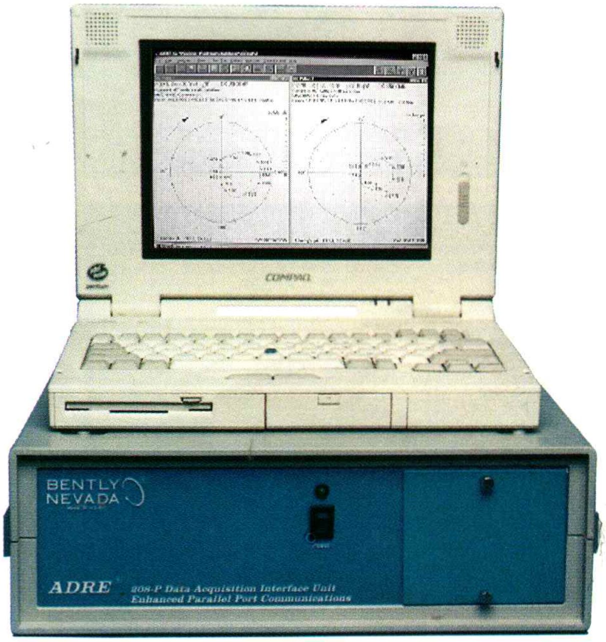
It is now easier to analyze ADRE for Windows' polar-format plots, because the plots can be quickly rescaled in predefined steps with one click of the mouse button.

Polar-format plots are used extensively in machinery diagnostics as they emphasize certain machine characteristics conveyed in the phase lag and amplitude of vibration components. A polar trend plot of data from a machine running at constant speed can identify deterioration and stress, often earlier than a trend of vibration amplitude alone. Polar-format plots of data captured from startups and shutdowns reveal fundamental machine characteristics that are essential for balancing, baseline comparison, and troubleshooting.

ADRE for Windows makes it easier to compare polar-format plots with other plots, and to show detail in the plots. With a single click of the mouse button, you can now rescale polar plots by a factor you select.

Better indication of "split resonances"

Lateral balance resonances are one of the fundamental machine characteristics effectively identified in transient data plots. However, the exact synchronous response, phase lag, and frequency characteristics of a resonance can be difficult to determine if the machine has a split resonance, and the transducers are not aligned with the stiff and weak axes of the machine. ADRE for Windows now makes it easier to determine information on machines with split resonances.


COMPAQ
BENTLY NEVADA
ADRE
208-P Data Acquisition Interface Unit
Enhanced Parallel Port Communications

ADRE® for Windows has become more versatile and more powerful. It now runs on many more types of notebook computers. The new 208-P Data Acquisition Interface Unit is shown here with a laptop computer.

When the transducers do not coincide with the stiffness axes, the plots contain information affected by both axes and are difficult to interpret. This phenomenon is more common in horizontal machines, because most horizontal machines are usually stiffer vertically than horizontally. ADRE for Windows can now rotate the transducers in its software to make these fundamental characteristics easier to measure. After the ADRE for Windows program rotates the transducers in the software to coincide with the weak and stiff axes, the information is far easier to interpret.

ADRE for Windows now lets you more accurately identify the resonance frequency, synchronous response, phase lag, and Synchronous Amplification Factor for the weak and stiff axes. The effect of a split resonance on a Bode and polar plot presentation can be observed. This technique was described in the June 1995 issue of the Orbit in an article entitled, “How to handle (and some cases of) forward and reverse orbits.” A copy of this article is available upon request by checking the appropriate box on the Reader Service Card in this issue.

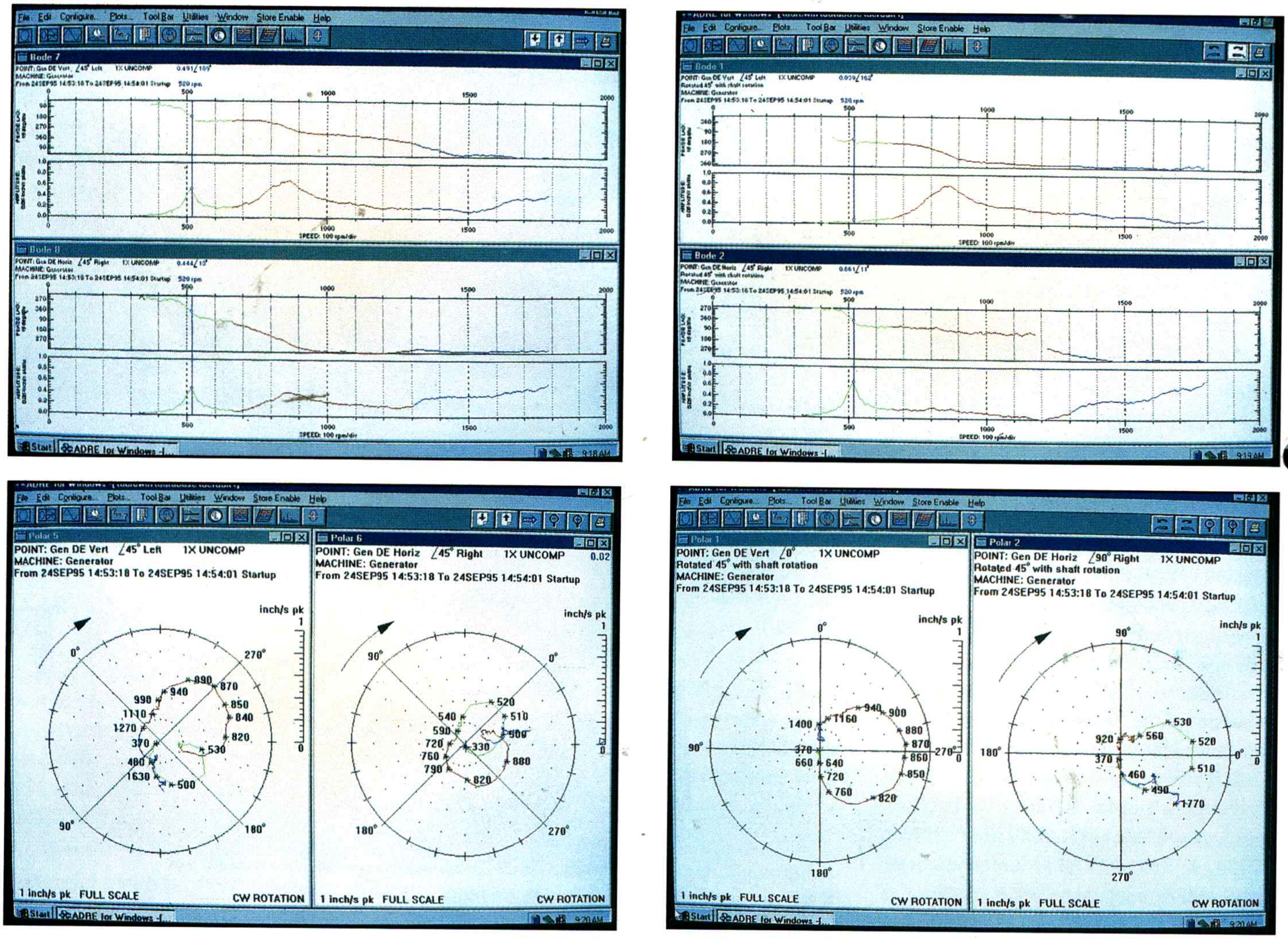
Improved documentation for spectrum plots

Spectrum plot documentation has been enhanced, because ADRE for Windows now lets you place up to 12 labels on each spectrum plot. The labels identify vibration components with frequency and amplitude information. Spectrum plot labels help you more easily identify the frequency and amplitude of a specific component.

World’s most powerful portable diagnostic instrument

Bently Nevada’s ADRE for Windows diagnostic software has become enormously popular because of its value. It is powerful; data from several points can be viewed simultaneously, each in its own window. It is flexible; data can be viewed in any of several plot formats, so the expert can choose the presentation that best exposes the parameter of interest. It is easy to use, portable, and battery-powered.

ADRE for Windows gives you essential information that makes problem diagnosis fast and accurate. Now, it is an even better value. Contact your nearest representative for more information and a complete demonstration. ■


Polar 5
POINT: Gen DE Vert ∠45° Left 1X UNCOMP
MACHINE: Generator
From 24SEP95 14:53:18 To 24SEP95 14:54:01 Startup

Polar 6
POINT: Gen DE Horiz ∠45° Right 1X UNCOMP 0.02
MACHINE: Generator
From 24SEP95 14:53:18 To 24SEP95 14:54:01 Startup

Polar 1
POINT: Gen DE Vert ∠0° 1X UNCOMP
Rotated 45° with shaft rotation
MACHINE: Generator
From 24SEP95 14:53:18 To 24SEP95 14:54:01 Startup

Polar 2
POINT: Gen DE Horiz ∠90° Right 1X UNCOMP
Rotated 45° with shaft rotation
MACHINE: Generator
From 24SEP95 14:53:18 To 24SEP95 14:54:01 Startup

Upper left is an ADRE® for Windows Bode plot *without* probe rotation. Upper right is the same Bode plot *with* probe rotation. Corresponding polar plot in lower left is *without* probe rotation. Lower right is the same polar plot *with* probe rotation.

New Trendmaster® 2000 flexiTIM™: simplified design significantly reduces cost

Ideal for expansion—100% compatible with existing installations

by Marco Alcalde
Product Marketing Manager
Small Machinery Solutions
Bently Nevada Corporation

Trendmaster® 2000 is a proven, high-value hardware and software solution for machinery data management of general-purpose and balance-of-plant rotating machinery. Its worldwide users have been able to reduce plant-wide maintenance expenses, as well as increase asset availability. Now, we have made the Trendmaster 2000 System even better. A redesigned Transducer Interface Module (TIM) provides greater mechanical protection and reduced installation complexity. The new, two-channel flexiTIM™ installs next to the machine on a 1-inch conduit junction box and uses rugged transducer cables with integral connectors and screw-type terminal blocks for the system cable. This new module simplifies cable and conduit installation, which reduces the cost of installation, expansion, and life cycle support. In many situations, the flexiTIM will help overcome the challenging hurdle of initial system justification. It is also ideal for expanding your current system, as it is 100% compatible with existing installations.

Business opportunity

In today's industry, the focus has shifted from process control optimization to business optimization. Effective rotating machinery asset management requires a clear understanding of asset condition, so maintenance can be efficiently planned. It also requires a clear understanding of the factors which limit the machine's service life and performance.

Through the automation of data acquisition, reliable asset and process information can be correlated and provided to both operations and reliability personnel 24 hours a day, 365 days a year. Problems are detected before significant damage occurs, allowing people to manage situations as well as identify and solve the fundamental problem.

Trendmaster® 2000 system

The Trendmaster® 2000 system consists of two layers. The hardware layer is a permanently-installed "sensor bus," which includes transducers and transducer interface modules (TIMs), and a PC data acquisition card installed in the Data Acquisition computer. A cable runs from the PC card to TIMs mounted on housings located near the machinery. Cables from the TIMs connect to transducers mounted on the machinery. This enables a single cable to support up to 255 TIM/Transducer combinations (65 in an Intrinsically Safe installation).

The software layer consists of the data acquisition, display, configuration, and Modbus®* and NetDDE server modules. The data acquisition module, along with the Modbus and NetDDE server modules, runs on the Data Acquisition computer under the Microsoft Windows NT™ operating system. The display and configuration modules will run on the Data Acquisition computer, and computers connected to it by Local and Wide Area Networks or remote access via telephone. These modules will run on any of the Microsoft operating systems. The first flexiTIM available will be a two-channel acceleration-to-velocity version.

Contact your local sales representative for more information.■

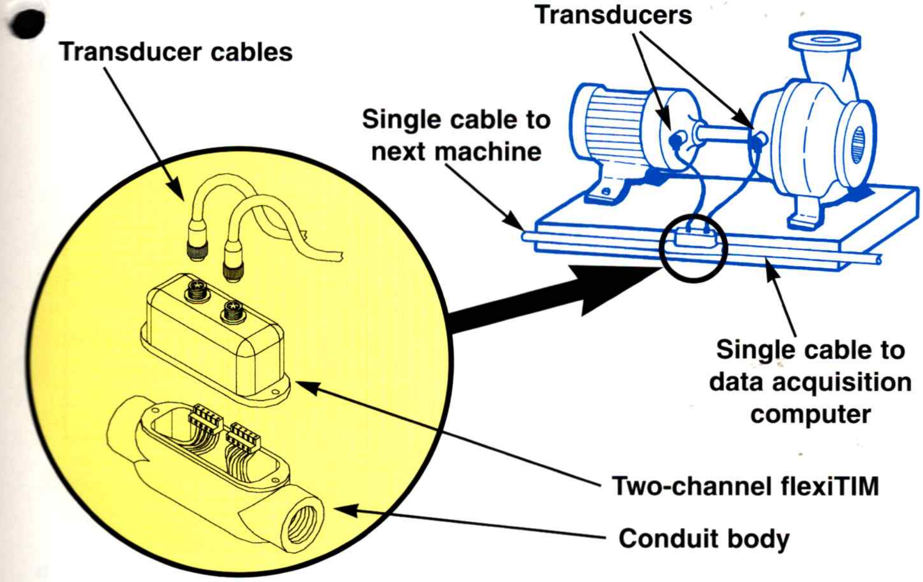

Transducer cables
Transducers
Single cable to next machine
Single cable to data acquisition computer
Two-channel flexiTIM
Conduit body

Figure 1
Detail drawing of the new two-channel flexiTIM™ for Trendmaster® 2000 showing a typical installation on a motor/pump set.

**Modbus® is a registered trademark of Modicon, Inc.*

Back-to-Basics

Why install a Keyphasor® transducer?

by Pat Russell
Northeast Region Manager
Bently Nevada Corporation

Vibration monitoring and diagnostics systems have three purposes. First, the system should alert Operators to developing problems, while they are still manageable. Second, to minimize damage to the machine, the system should provide shutdown capability in the event of a catastrophic failure. Third, and perhaps most important, it should provide information to help the Machinery Specialist quickly and accurately diagnose machinery conditions.

The Keyphasor transducer is an essential element of a monitoring and diagnostics system because it provides both the Operator and Machinery Specialist with information vital to the successful protection and management of plant machinery. The Keyphasor signal, (Figure 2) a once-per-turn voltage pulse provided by a transducer (normally a

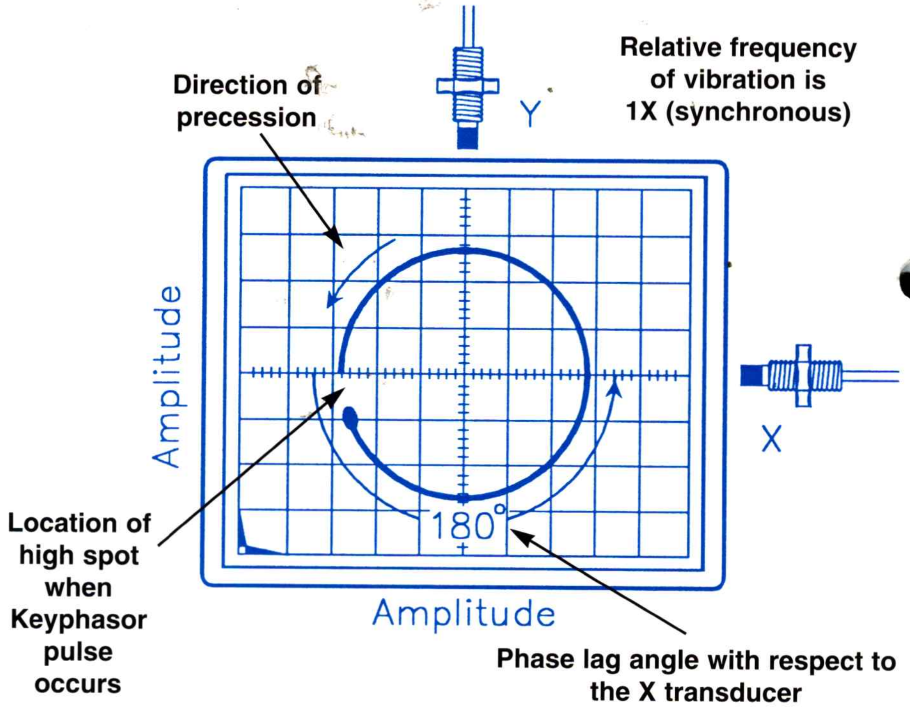

Direction of precession
Y
Relative frequency of vibration is 1X (synchronous)
Amplitude
X
Location of high spot when Keyphasor pulse occurs
180°
Amplitude
Phase lag angle with respect to the X transducer

Figure 1
Two (XY) proximity transducers produce a two-dimensional picture of shaft motion in the measurement plane. By examining the Keyphasor mark (the sequence of blank/bright spots) on the orbit, the direction of precession (forward or reverse) can then be determined.

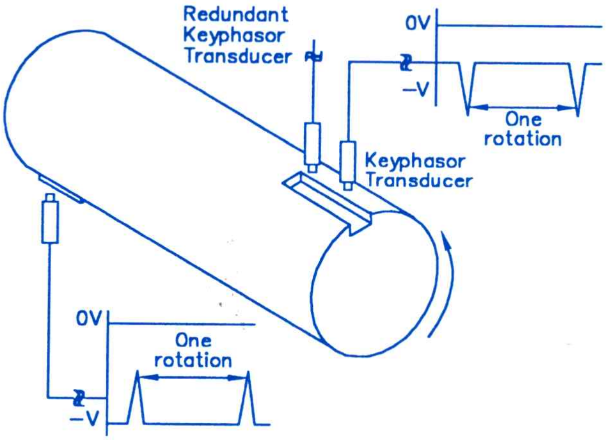

Redundant Keyphasor Transducer
0V
−V
One rotation
Keyphasor Transducer
0V
One rotation
−V

Figure 2
The Keyphasor once-per-turn reference signal can come from a variety of sources: A proximity probe observing a notch or projection on the shaft, an optical pickup observing reflective tape, or a strobe light trigger using a match mark on the shaft and casing.

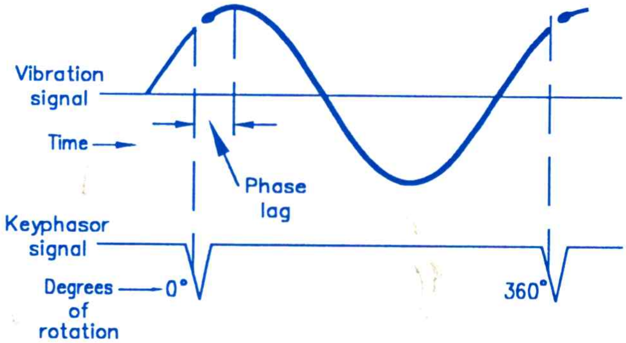


Figure 3
Absolute phase is the number of degrees of vibration cycle from when the Keyphasor fires to the first positive peak in the vibration signal. It is always a lag angle, measured from 0 to 360 degrees.

proximity probe), is used by the monitoring system to generate filtered vibration amplitude and phase lag measurements. The Keyphasor information can help the Operator identify developing machine problems or distinguish serious problems from less serious ones.

Information for the Operator

In the past, Operators were only required to manage the process. In today's competitive environment, they are expected to manage not only the process, but plant assets, including machinery. In some plants, the Operator can call a Machinery Specialist to analyze machinery data and determine what should be done. However, in many plants, a specialist may not be immediately available. If the Operator only has direct (or overall) vibration amplitude measurements, it is difficult to decide whether the machinery problem is serious or not.

A permanently-installed Keyphasor transducer provides the Operator with phase information which can help him detect subtle changes to the machinery. Operators can use the Bently Nevada 3500 Machinery Data Management System, the Trendmaster® 2000 System or the Data Manager® 2000 System for Windows NT™ to generate industry-standard plots that display 1X (synchronous) amplitude and phase or NOT 1X (nonsynchronous) amplitude measurements. These Bently Nevada systems were designed to present 1X and NOT 1X information in a way that is easy for Operators to measure. Operators can easily compare the 1X portion of the vibration signal to the NOT 1X portion.

Changes in 1X vibration need to be analyzed. If the change is gradual, the Operator can schedule a visit by the Machinery Specialist. Then, the operator can continue to manage his machinery and process with increased attention. If the change is sudden, an immediate call to the specialist is in order, and the Operator can apply the appropriate management procedure to this new situation.

Serious machine problems

A change in the NOT 1X portion of the overall vibration, however, is often a strong indicator of malfunctions that can quickly be detrimental to the machine. If there has been an increase in NOT 1X vibration, a Machinery Specialist should be called immediately. Some of the most common causes of NOT 1X vibration are fluid-induced instabilities, rotor-to-stator rubs, and gear faults. All are of immediate concern.

Information for the Machinery Specialist

The Machinery Specialist also needs machinery information from a Keyphasor transducer to perform his job properly. Although some plot formats, such as shaft orbits and spectrum displays, will be available, a Keyphasor mark on orbit plots (Figure 1) is necessary to evaluate phase lag. Without the Keyphasor mark, the shaft orbit will not indicate if the vibration is occurring in a synchronous (1X) or nonsynchronous mode. A Keyphasor mark is also necessary to determine the direction of shaft vibration precession (in the same or the opposite direction of shaft rotation) from the orbit, and to generate the mode shape of the shaft (Figure 4).

In addition, polar and Bode plots, which are necessary to document rotor response on startup or shutdown, cannot be developed without Keyphasor information. These plots are also needed for accurate balancing. Before any balancing solutions are attempted, polar plots should be reviewed to verify where the weights should be placed.

The Machinery Specialist needs to review as much machinery data as possible to provide an accurate analysis. The lack of Keyphasor-related data can lead to an incorrect diagnosis.

It is sometimes difficult to install a Keyphasor transducer. However, the additional information provided by the Keyphasor signal is absolutely necessary for Operators and Machinery Specialists to successfully protect and manage machinery. For this reason, both Bently Nevada and the American Petroleum Institute, in its specifications for machinery protection systems, recommend installing a Keyphasor transducer and a redundant Keyphasor transducer. ■

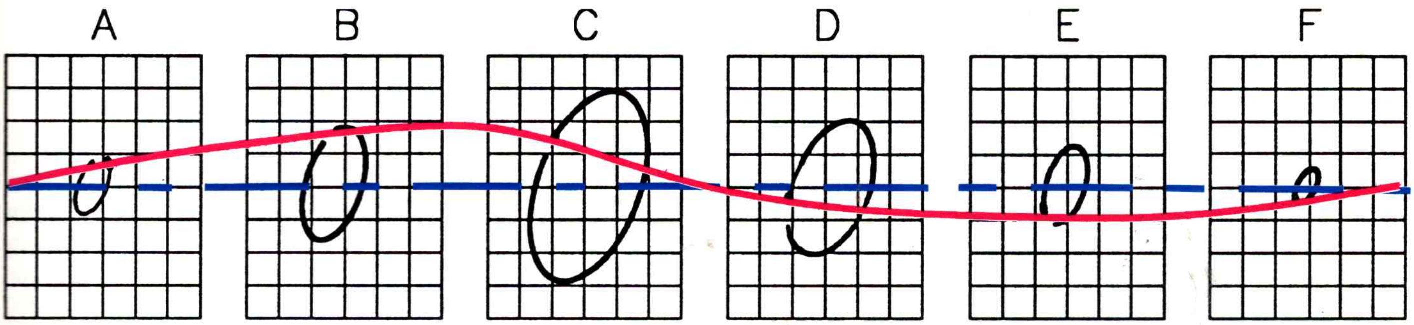


Figure 4
Example of mode shape

A through F above are orbits shown at the same speed at different bearings. The blue dashed line represents the orbit centerline. The red line connecting the Keyphasor dots shows the apparent shaft deflection shape.

Bently Nevada Product Service:

Serving our customers with expertly trained personnel

by ***Tom Pfoh***
Corporate Product Service Manager
Bently Nevada Corporation

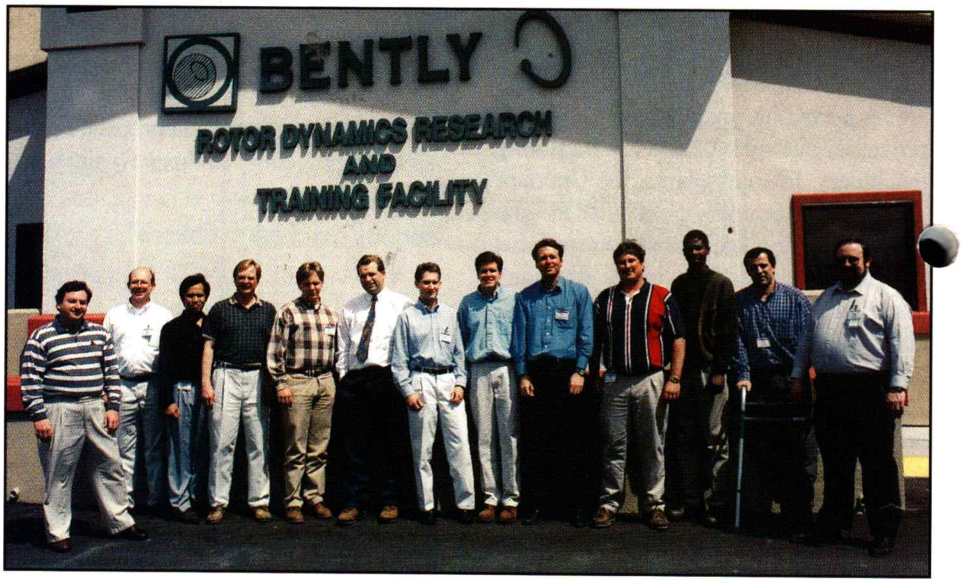
Two years ago, Bently Nevada developed a program to formalize training for both entry level and experienced Product Service personnel. Each group attends an intensive, seven-week program that includes safety training and customer relationship skill development.

Our newly-hired personnel also learn about the philosophy and history of Bently Nevada. They receive extensive training in installation, calibration, operation, and troubleshooting of all our products, old and new. They also learn basic machinery diagnostics, so they can understand the machinery and the purpose that our systems serve.

Our more experienced service people take time to refresh their understanding of machinery diagnostics. They expand their skills in the areas of digital communications and Windows NT™. Both of these areas are vital to Bently Nevada's objective of moving data, not people. They spend the balance of their time learning the installation and operation of our latest software.

Because many of our customers now connect our products to their control systems and plant information networks, we need people with more specialized training. Selected service personnel receive eight weeks of advanced training in computer operating systems and applications, and local and wide area networking. Over the last two years, more than a dozen of our people have completed the program presented by a company specializing in this area.

We are committed to providing our customers with the most qualified Product Service personnel available.■



Some of our more experienced service personnel at a recent training program. Left to right: Billy Gilkerson, Instructor; Tim Tilger, Raden Bashirudin, Jim White, Jorgen Jansson, Andrew Bibby, Bobby Stodghill, Rod Fierbach, Eddie Cafferty, Tim Gross, Mike Thomas, Mazen Kashou, Chris Deaton, Instructor; and Rick Martell (not present).

Design & Installation Services: At your service

Safety is an important part of our training. Design and Installation Services personnel have retrofitted over 300 different machine trains without a startup delay or lost time injury.

by ***Mark Colliflower***
Design and Installation
Services Manager
Bently Nevada Corporation

Bently Nevada formed Turnkey Services fourteen years ago. This group is specifically dedicated to providing customers with high quality, complete turnkey installations of our machinery protection and information systems.

Over the years, the services offered have changed from exclusively performing turnkey installations to performing individual aspects of a complete installation. The group's name was changed to Design & Installation Services (D&IS) to reflect those increased capabilities. We have performed jobs consisting of portions of overall projects. These have included project management, electrical design and installation, mechanical design and installation, system setup, configuration, commissioning, and documentation. We have found, however, that the highest quality and lowest cost installations are turnkey installations.

Our customers have used D&IS for a number of reasons. They didn't have the time or people to learn to apply our products to their machinery, develop electrical and mechanical designs, and define and purchase required materials. They weren't able to use those materials to build the brackets and do the machining necessary to complete the installation. They might not have had the time or personnel to completely and accurately document all the work, documentation that would allow them to maintain the plant efficiently and comply with regulations that governed plant changes. D&IS takes care of all these aspects of an installation project.

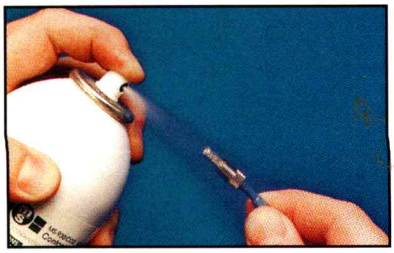
D&IS personnel typically hold degrees in Mechanical Engineering. They undergo an extensive Bently Nevada Product Service training program as well as a year of on-the-job training under the guidance of our Senior Site Supervisors. Safety training is one of the most important parts of the program. As a result, our personnel have retrofitted over 300 different machine trains without a startup delay and without a lost time injury.

All of our individual installation designs undergo an internal review for correct dimensioning and proper application (using our knowledge-base of past applications and machine installations). From this review process, we continually refine our designs. We apply feedback from our Machinery Diagnostic Services group and from the Bently Rotor Dynamics Research Corporation to each new installation. This means that we supply our customers with the most up-to-date capabilities for protecting and managing their machines.

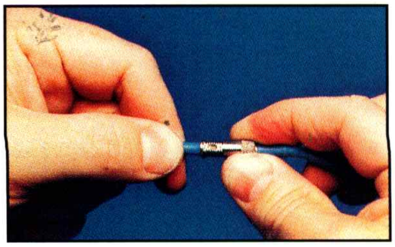
Contact your local representative to discuss Design & Installation Services' installation of your next machinery protection and management system. ■

ProbeTip

Connectors: Keep them snug and secure



1. Clean the connector of dirt and grease by spraying it with an electrical contact cleaner.



2. Connect the two halves by sliding back the ferrule and pushing the two pieces together. Tighten the ferrule of the connector finger tight.

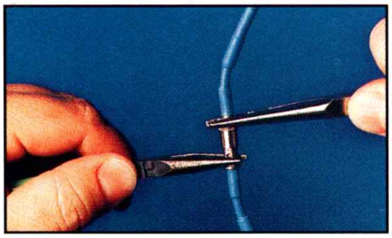
One of the weakest links in any electrical system is the connection point. Over the years, we have seen instances where intermittent readings, alarms, and even false shutdowns, have been traced to improperly made connections and improperly assembled connectors.

To reduce problems with connectors, we introduced our 3300 Series Proximity Transducer System several years ago with connectors made from stainless steel. This material has been found to be more robust and resistant to mechanical damage than the silver-plated brass used previously. If not installed properly, however, it can loosen and become a weak link in the system.

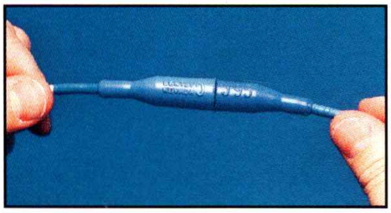
These five illustrated steps can help you make a good connection with your 3300 Series transducers.

Use of a Bently Nevada Connector Protector is recommended because they are reusable, provide excellent electrical isolation from ground, and are resistant to almost all chemicals.

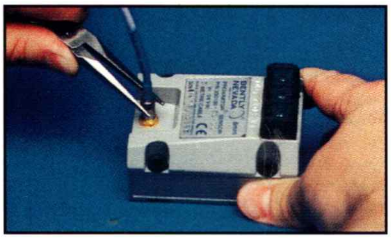
Paying a little extra attention to the connections on your transducer system will make sure that your machinery protection and management system gives you reliable service. ■



3. Tighten the ferrule another 1/8 turn with pliers. This will ensure that the stainless steel surfaces will not disengage.



4. Cover the connector with a Bently Nevada Connector Protector, self-fusing silicone tape, or heat shrink tubing.



5. Remember to also tighten the connector on the Proximitor® finger tight plus 1/8 turn more.

1997 International Seminar Schedule

DATE	COURSE DESCRIPTION	COUNTRY	LOCATION
June 2-6	Machinery Diagnostics	Asia	Kuala Lumpur, Malaysia
June 9-12	Machinery Diagnostics Workshop		Kuala Lumpur, Malaysia
December 2-5	Machinery Diagnostics	Australia	Sydney
July 7-11	Applied Diagnostics	Canada	Canmore, Alberta
July 21-25	Applied Diagnostics		Muskokas, Ontario
August 25-28	Survey of Machinery Monitoring		Sudbury, Ontario
September 8-12	Machinery Diagnostics	England	Warrington
September 15-19	Applied Diagnostics		Warrington
September 22-26	3500 Installation & Maintenance		Warrington*
October 7-9	Machinery Monitoring		Warrington
October 14-16	3300 Installation & Maintenance		Warrington*
October 14-17	Data Acquisition	France	Nantes
October 21-24	Machinery Diagnostics	Germany	Neu-Isenburg / Frankfurt
October 27-30	Applied Diagnostics		Neu-Isenburg / Frankfurt**
November 3	Transducer Installation & Maintenance		Neu-Isenburg / Frankfurt
November 4-5	3300 Installation & Maintenance		Neu-Isenburg / Frankfurt*
October 21-24	Data Acquisition and Machinery Monitoring	Korea	Seoul
October 28-31	Machinery Diagnostics		Seoul
September 8-12	Data Acquisition	The Netherlands	Pijnacker
September 22-26	Survey of Machinery Monitoring		Pijnacker
September 30 - October 2	3300 Installation & Maintenance		Pijnacker*
October 27-31	Getting the Most from Data Manager 2000 for Windows		Pijnacker
November 18-20	Getting the Most from ADRE for Windows		Pijnacker
November 24-28	Machinery Diagnostics		Pijnacker
December 9-11	3500 Installation and Maintenance		Pijnacker*
November 5-7	Survey of Machinery Monitoring	Norway	Bergen
November 11-14	Machinery Diagnostics		Bergen**
June 14-18	Machinery Diagnostics	Saudi Arabia	Dammam**
June 21-25	Applied Diagnostics		Dammam**
September 13-17	3300 Installation & Maintenance		Dammam** *
September 21-24	3300 Installation & Maintenance		Dammam** *
September 27-October 1	3500 Installation & Maintenance		Dammam** *
October 4-8	Machinery Diagnostics		Dammam**
October 11-15	Machinery Diagnostics		Dammam**
October 18-22	Applied Diagnostics		Dammam**
October 13-15	Survey of Machinery Monitoring	Sweden	Stockholm
August 18-22	Machinery Diagnostics	Thailand	Map Ta Phut, Rayong Prov.
November 3-7	Machinery Diagnostics		Map Ta Phut, Rayong Prov.
June 4-6	Getting the Most from ADRE for Windows	United States	Philadelphia, Pennsylvania
June 9-13	Machinery Diagnostics		Philadelphia, Pennsylvania
June 10-12	3300 Installation & Maintenance		Houston, Texas*
June 16-20	Machinery Diagnostics		Minden, Nevada
June 16-20	Applied Diagnostics		Philadelphia, Pennsylvania
June 17-19	Getting the Most from Data Manager 2000 for Windows		Houston, Texas
June 23-27	Applied Diagnostics		Minden, Nevada
July 22-24	Getting the Most from Data Manager 2000 for Windows		Minden, Nevada
July 22-24	3300 Installation & Maintenance		Philadelphia, Pennsylvania*
July 23-25	Getting the Most from ADRE for Windows		Houston, Texas
July 29-31	Getting the Most from Data Manager 2000 for Windows		Philadelphia, Pennsylvania
July 28- August 1	Advanced Machinery Dynamics		Minden, Nevada
July 28- August 1	Machinery Diagnostics		Houston, Texas
August 4-8	Applied Diagnostics		Houston, Texas
August 5-7	Getting the Most from Trendmaster 2000 for Windows		Philadelphia, Pennsylvania
August 11-15	Machinery Diagnostics		Minden, Nevada
August 13-15	Transducer Installation & Maintenance		Houston, Texas
August 18-22	Applied Diagnostics		Minden, Nevada
August 19-21	3300 Installation & Maintenance		Houston, Texas*
August 26-28	Getting the Most from Data Manager 2000 for Windows		Houston, Texas
September 9-11	3300 Installation & Maintenance		Minden, Nevada*
September 16-18	Getting the Most from Trendmaster 2000 for Windows		Minden, Nevada
September 23-25	3500 Installation & Maintenance		Houston, Texas*
September 30-October 2	Transducer Installation & Maintenance		Philadelphia, Pennsylvania
October 7-9	3500 Installation & Maintenance		Philadelphia, Pennsylvania*
October 7-9	Getting the Most from Trendmaster 2000 for Windows		Houston, Texas
October 14-16	Getting the Most from Data Manager 2000 for Windows		Philadelphia, Pennsylvania
October 22-24	Getting the Most from ADRE for Windows		Houston, Texas
October 27-31	Machinery Diagnostics		Houston, Texas
November 3-7	Applied Diagnostics		Houston, Texas
November 5-7	Getting the Most from ADRE for Windows		Philadelphia, Pennsylvania
November 10-14	Machinery Diagnostics		Philadelphia, Pennsylvania
November 11-13	3300 Installation & Maintenance		Houston, Texas*
November 17-21	Applied Diagnostics		Philadelphia, Pennsylvania
November 18-20	Getting the Most from Data Manager 2000 for Windows		Houston, Texas

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*** Optional one extra day for additional topics. ** This seminar is conducted in English.**

Seminars are usually conducted in the language customarily spoken in the host country.

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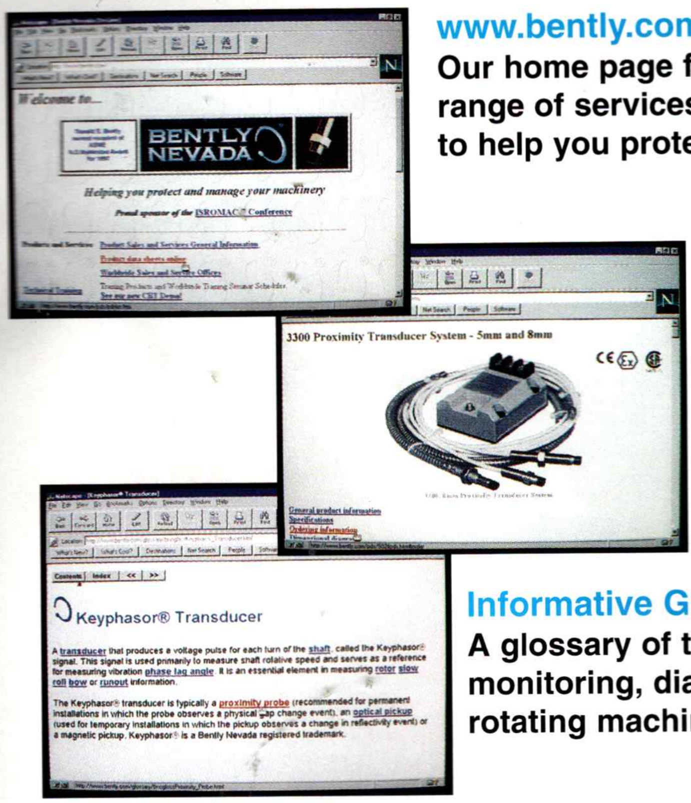

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A Technical Publication

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1617 Water Street
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